MAY 4 1931

Heating · Piping and Air Conditioning

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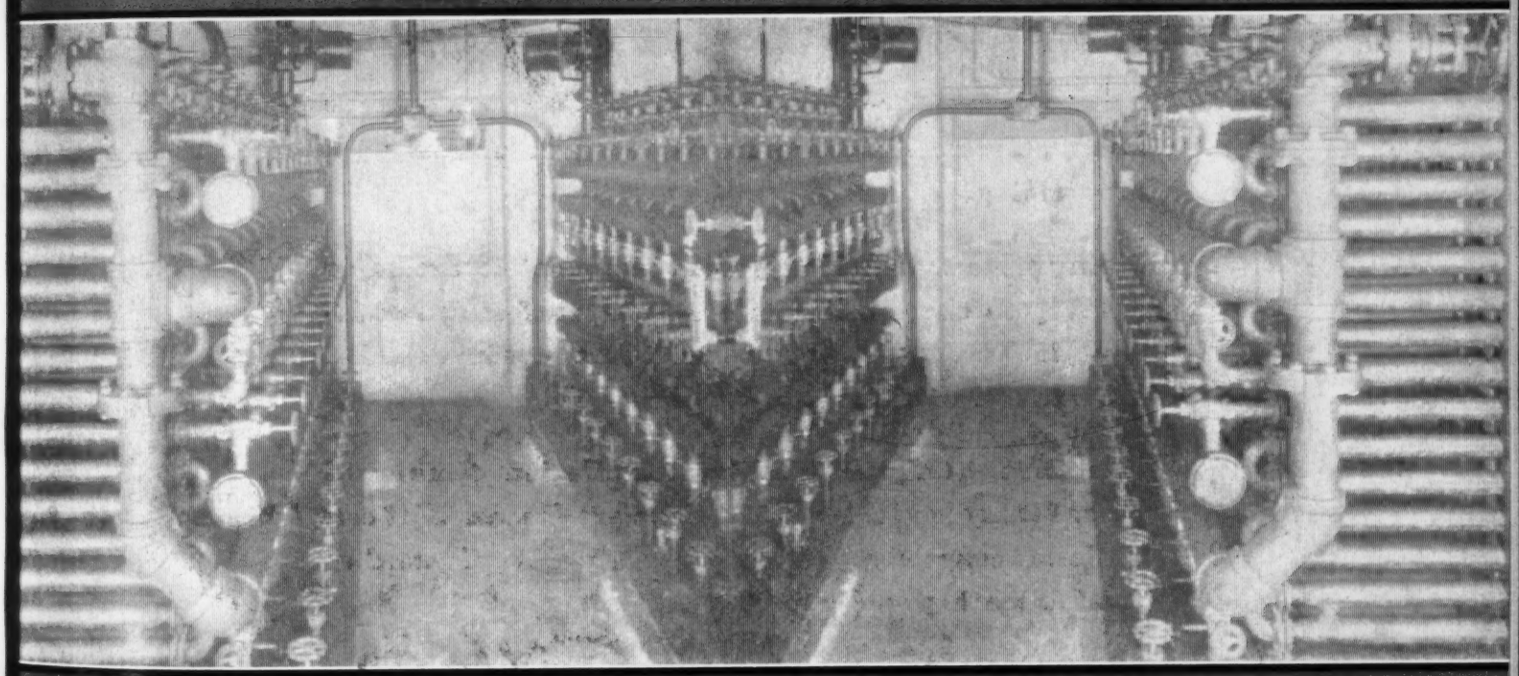
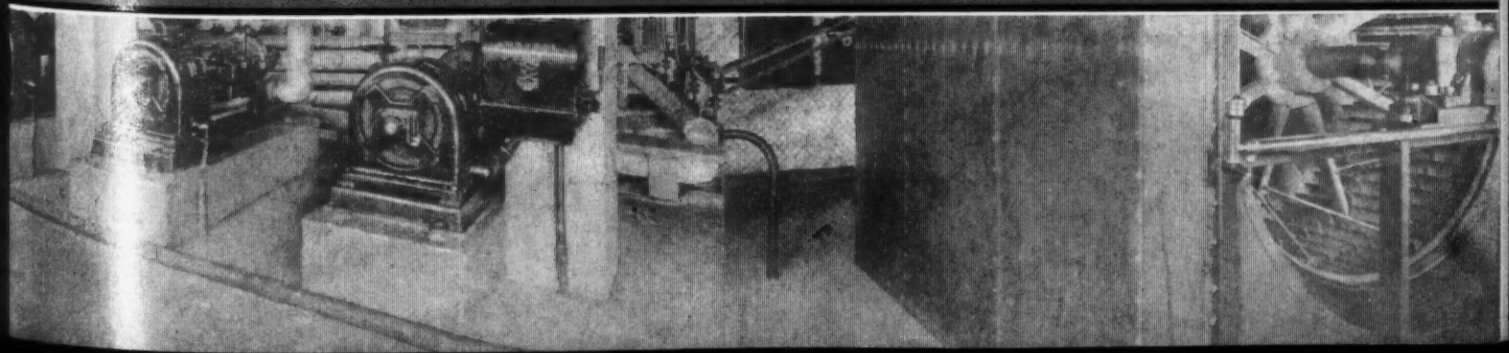
JOURNAL SECTION OF THE A.S.H.V.E.

MAY, 1931



AN ARTICLE BY LEE P. HYNES ON HANDLING VISCOUS MATERIALS IN PIPING—TURN TO PAGE 363 » » » »



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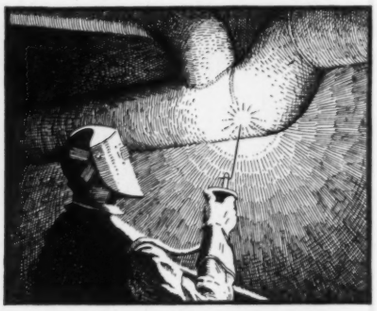

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The EDITOR'S PAGE

Heating · Piping and Air Conditioning

FROM ARTICLES IN THIS ISSUE

"There are many methods of heating and reheating, including steam, superheated steam, direct firing, and electricity; all have their proper applications and require careful design, erection and operation . . . we will deal with a specific case . . ." —*Lee P. Hynes.*

"What attention should be given to the air filter for it to function without interruption, with entire satisfaction, and with maximum efficiency?"—*W. G. Frank.*

"*Kal.* This means the steam which ensues when we evaporate 1000 lb. of water. We do need an accredited term meaning a larger unit than a pound in describing a quantity of steam." —*"Open for Discussion."*

"In piping engineering, as well as in every other branch of human activity, all goes well until man gets the idea that he knows more than Mother Nature, and starts to high hat her. Then the fun begins."—*Henry G. Schaefer.*

"As a means of eliminating this trouble, and keeping the vicinity immediately surrounding the laundry free from lint, the lint collector shown in the sketch may be constructed."—*W. H. Pierce.*

"An 8-in. C. W. line runs from the distributing header to the air conditioning apparatus. The air conditioning equipment, when operating at 100 per cent capacity, will use approximately 750 g. p. m. for condensing purposes. This quantity closely approximates the demand of the building for the fixture supply."—*Staff of Clyde R. Place.*

"It is not always necessary to spend a large amount of money for metering equipment. Arrangements can usually be made, by having suitable piping connections installed, so that a meter can be used in one place for part of the time and then moved to another location."—*W. H. Wilson.*

"Enough has been said to indicate the necessary steps for proper maintenance. Wear is constantly going on and leakage is constantly developing. . . . (Proper maintenance) can be accomplished only by a definite system of testing, inspection and replacement based on accurate records."—*John H. Ruckman.*

"It will be seen that the use of corrosion retarders (in refrigerating systems) is not always mandatory, but should be determined on the basis of cost of treatment versus savings afforded."—*R. C. Doremus.*

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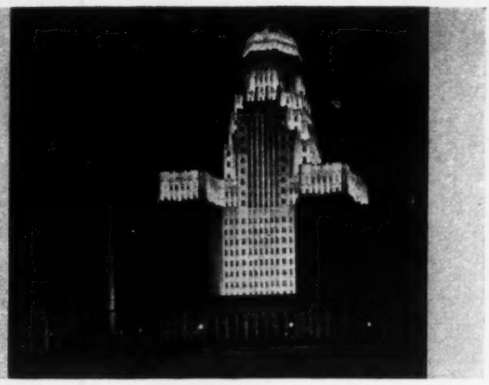

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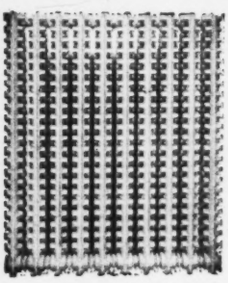

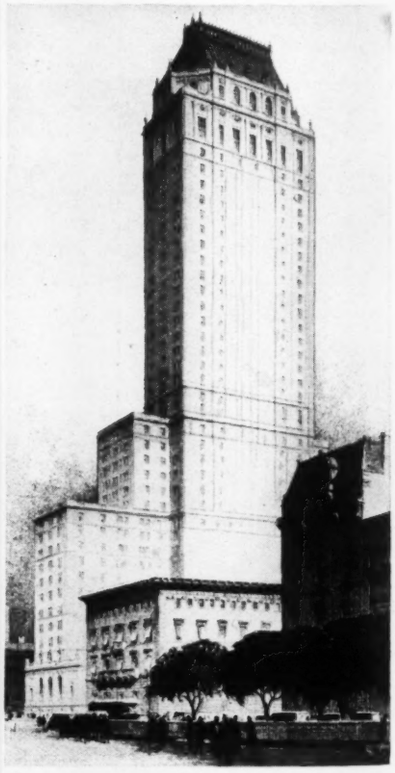

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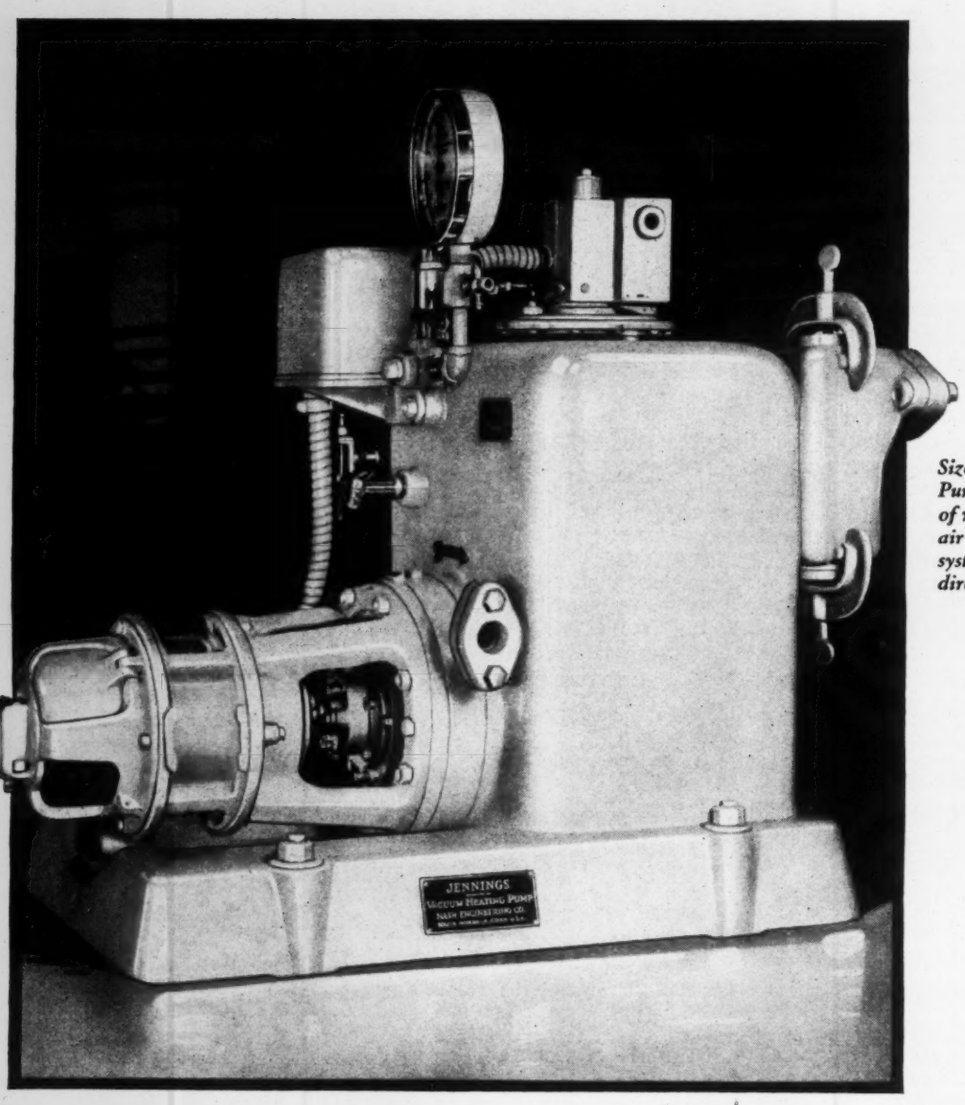

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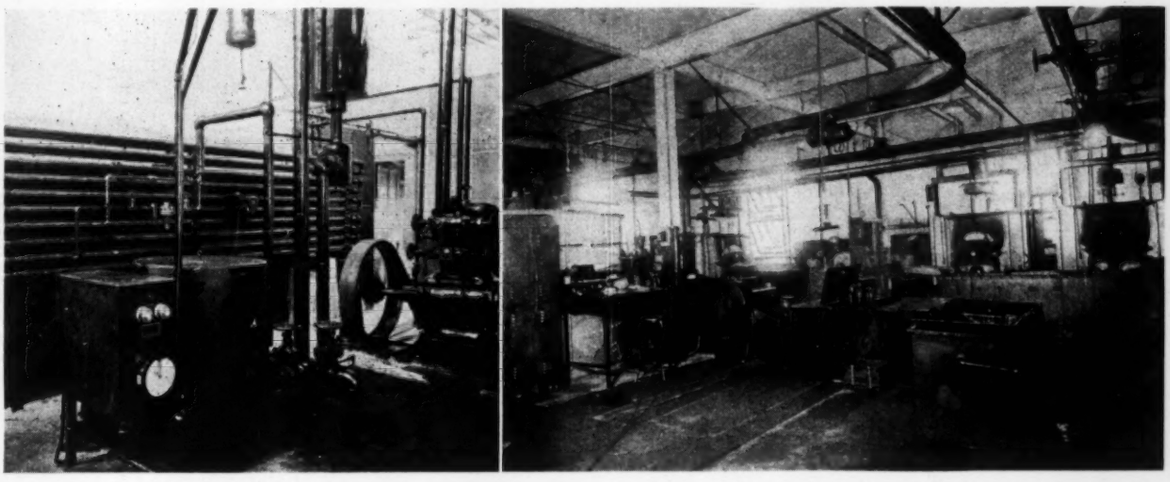


RIGHT—A HOT OIL TEMPERING PROCESS IN THE MANUFACTURE OF BALL BEARINGS. LEFT—THE HEATER FOR THE OIL. THE OIL IS PUMPED THROUGH PIPES LAID IN INSULATED FLOOR TROUGHS IN THE TREATING TANKS

Handling Viscous Materials

A Discussion of Some of the Factors That Must Be Considered to Insure a Successful Installation of the Piping

By Lee P. Hynes*

THROUGH the use of heat, it is possible to convey through pipes many materials which are very viscous or even entirely solid at ordinary temperatures; this article will deal principally with some of the special problems encountered in doing so. The subject naturally divides into two classes, one dealing with bulk materials which are heated simply for the purpose of making them fluid enough to transport from one point to another by pipe line. The other deals with materials which are heated and pumped, but where other aims are to be achieved besides the mere moving of the material.

We shall first discuss the former, where the primary object is to convey material more readily. In this class are heavy oils, greases, pitch, tar, waxes, syrups, and highly viscous compounds of many kinds. In handling these it is necessary to supply enough heat to lower the viscosity to a point where free flow can be insured at practical operating pressures and with economical power input at the pumps. Sometimes heat is needed only at the pump, but in other cases, part or all of the pipe line must be heated, or certain fittings and valves must be kept hot to reduce resistance to flow. In every case, care and experience are needed to design and install a satisfactory system.

The first essential is to determine the proper size and arrangement of pipe line capable of handling the desired volume of fluid at a practical temperature without undue friction losses. Naturally the pipe should be straight if possible, as every bend or fitting materially increases the friction loss or pressure drop. Wherever possible, long bends should be used as an ordinary elbow is equal in pressure drop to $\frac{(30 \times \text{diameter of pipe})}{12}$ feet of straight pipe. For a 6-in. line, this is equal to 15 ft. of

* Consulting engineer, Philadelphia, Pa.

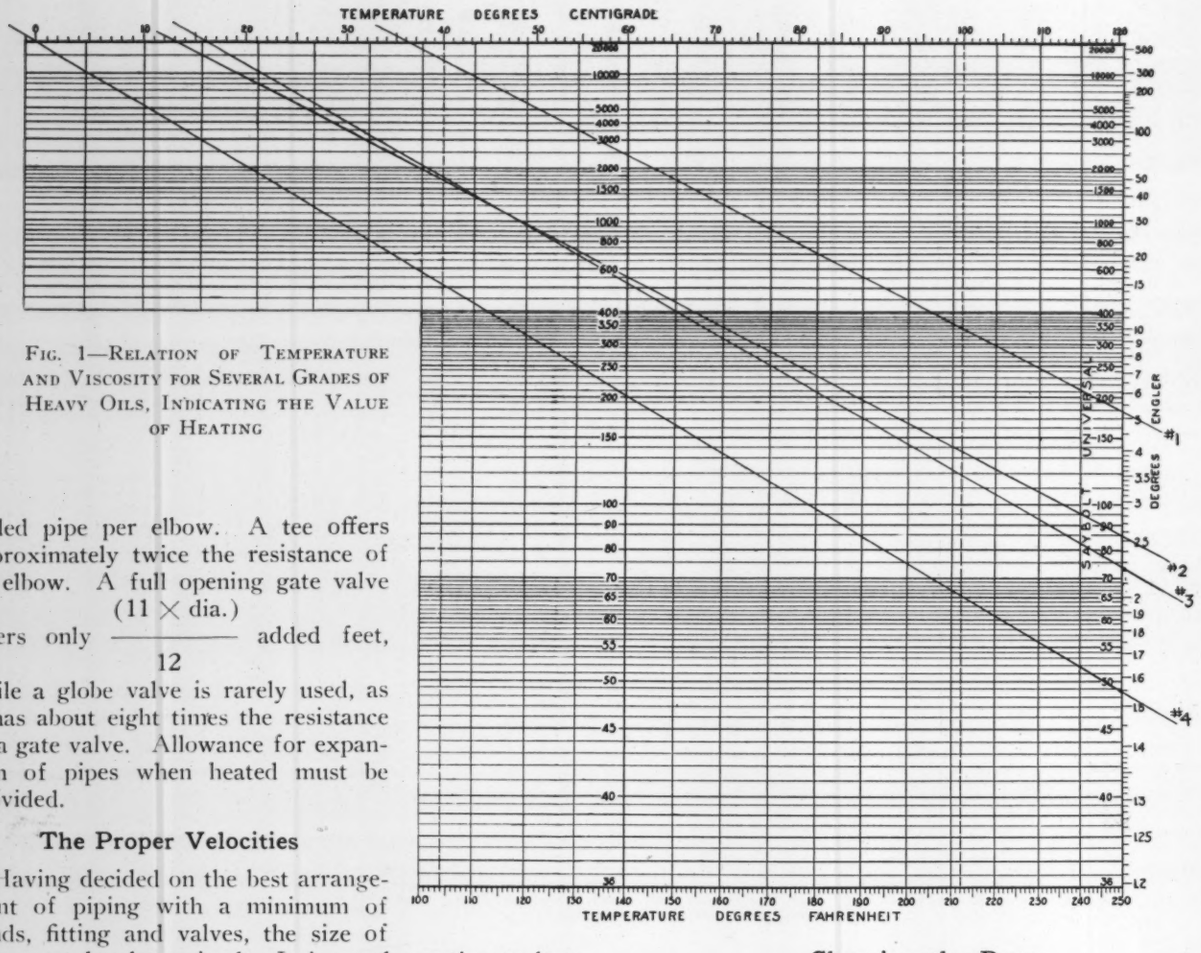


FIG. 1—RELATION OF TEMPERATURE AND VISCOSITY FOR SEVERAL GRADES OF HEAVY OILS, INDICATING THE VALUE OF HEATING

added pipe per elbow. A tee offers approximately twice the resistance of an elbow. A full opening gate valve offers only $\frac{(11 \times \text{dia.})}{12}$ added feet, while a globe valve is rarely used, as it has about eight times the resistance of a gate valve. Allowance for expansion of pipes when heated must be provided.

The Proper Velocities

Having decided on the best arrangement of piping with a minimum of bends, fitting and valves, the size of pipe must be determined. It is good practice to keep the velocity of flow as low as 3 ft. per sec. for suction lines; 5 ft. per sec. for short discharge lines; and not over 4 ft. per sec. for long discharge lines, in order to avoid turbulence.

The proper kind of flow in a pipe line is a smooth stream line flow, of continuous, concentric, cylindrical films of fluid parallel to the axis of the pipe. There is a critical velocity of flow where turbulence begins and instead of a smooth stream, the flow becomes a confused turbulent mass of eddies, which add greatly to the friction loss.

Pressure Drop

Having determined on a pipe diameter ample to allow a proper velocity of flow, and the equivalent straight pipe length of the system, the pressure drop in the pipe line must be considered, and by reference to Danforth's tables for oil or other standard formulas covering the material in question, the pressure drop for a given viscosity and gravity noted. As viscosity is the main factor in pressure drop, it will be obvious whether heating should be resorted to for reducing the viscosity to the most desirable point. The chart, Fig. 1, shows the relation of temperature and viscosity for several grades of heavy oils, and shows the value of heating.

Where steam is available, it is an excellent means for heating. Steam jacketed pumps, valves, fittings, and pipes can be used. Figs. 2 to 6 illustrate several pieces of steam jacketed equipment.

Choosing the Pump

After the pipe sizes, viscosity, temperature, gravity, and pressure drop of a system are all known, the pump and its power drive must be chosen. Positive action pumps are necessary which will pump against pressure and handle air, gas, or vapor as well as liquids.

It is particularly important that the suction of the pump be heated so that the pump can start properly and have power to force the fluid through the piping. A by-pass for the pump between discharge and suction is often desirable to reduce the starting load, and a pressure relief valve should be installed on the discharge side of pump to permit release of any excess pressure developed through obstruction to flow.

Figs. 7 to 10 show several pumping and piping installations handling heavy viscous fluids.

The Second Class of Problems

Let us now consider the second class of problems dealing with materials which are heated and pumped, not merely to transport them but for some other objective. Sometimes this is a step in some process of the manufacture of the material itself, as in the blending of soap stock, greases, and cosmetics, or in the purification of oils. Again the process may be one of transferring heat. Even a material as solid as lead can readily be melted and pumped through pipes to the required point. An example of this is in the manufacture of some types of rubber hose, in which a thin layer of lead is cast around the hose as it comes from the machine, as a part of the

manufacturing process, and is later stripped away and re-melted so that a continuous cycle using the same metal, is kept up.

In other cases, asphaltum compounds of various kinds, which are ordinarily in a solid form, are melted and dis-

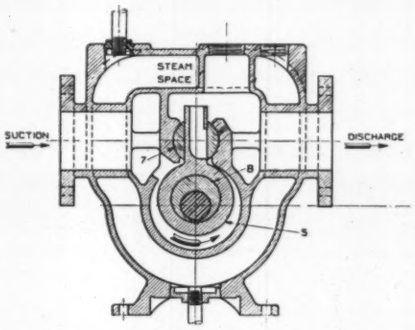


FIG. 2—A STEAM JACKETED PUMP

tributed for coating paper, metals, and other materials. In the extraction of grease and similar by-products from slaughter houses and packing plants, in soap works, and in many chemical industries, there are many materials of high viscosity which require special treatment if they are to be conveyed through pipes.

Each case requires a careful study of all the problems involved. These include proper pumps for handling the material, proper methods of heating, adequate heat insulation, and, frequently, re-heating to prevent the material from solidifying in the pipe lines or in valves or fittings. A viscosity greater than that for which the system has been designed will cause excessive pressure and overload the pumps, and may completely plug up the whole system.

There are many methods of heating and reheating, including steam, superheated steam, direct firing, and electricity; all have their proper applications and require careful design, erection, and operation.

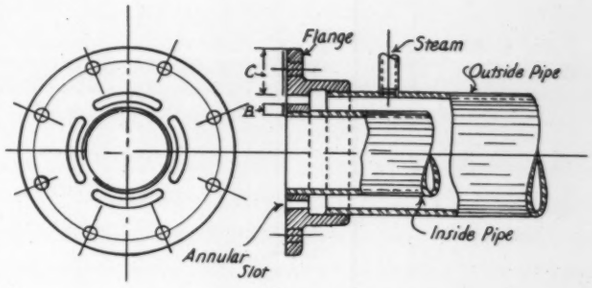
Heating Jacketed Vessels with Hot Oil

As each problem is a special one, we will deal with a specific case which will serve to illustrate how carefully it is necessary to study each particular installation if satisfactory results are to be secured. Consider the problem frequently met with in chemical plants, of heating jacketed vessels by pumping hot oil through them.

By this means, uniform heating may be obtained at temperatures higher than is possible with steam at permissible pressures. This is a heat transfer process wherein a suitable fluid medium is heated at one point of a closed circulation system and the heat extracted from it where required.

This is a relatively simple process for low temperatures but becomes much more difficult as the temperature increases because only liquids having high viscosity at ordinary temperatures will stand the higher temperatures, and there are difficulties in heating and pumping materials of high viscosity.

Heavy oils are now available which have been especially refined and tested for service of this kind and it is possible to heat satisfactorily some of these materials to 650 F under suitable precautions. The temperature and viscosity chart (Fig. 1) illustrates the change in



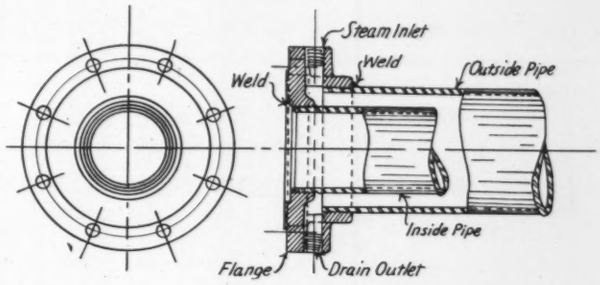


FIG. 3—CROSS-SECTIONS OF STEAM JACKETED PIPE

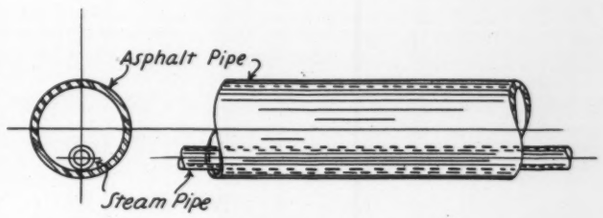


FIG. 4—PIPE LINE WARMED BY INTERNAL STEAM PIPE

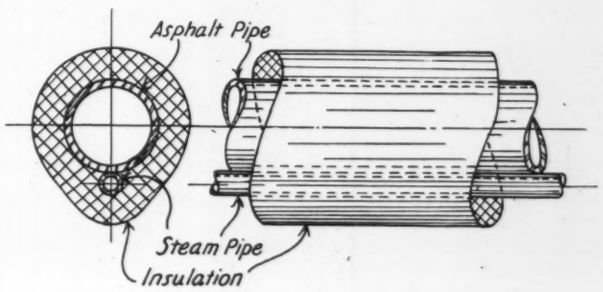


FIG. 5—PIPE LINE WARMED BY EXTERNAL STEAM PIPE

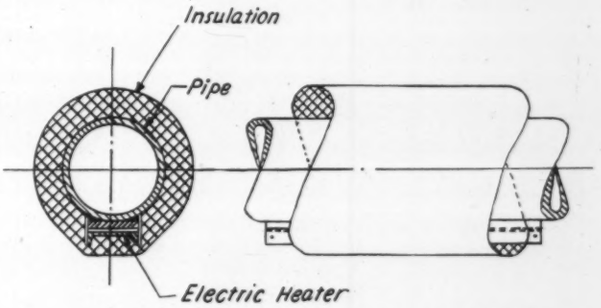
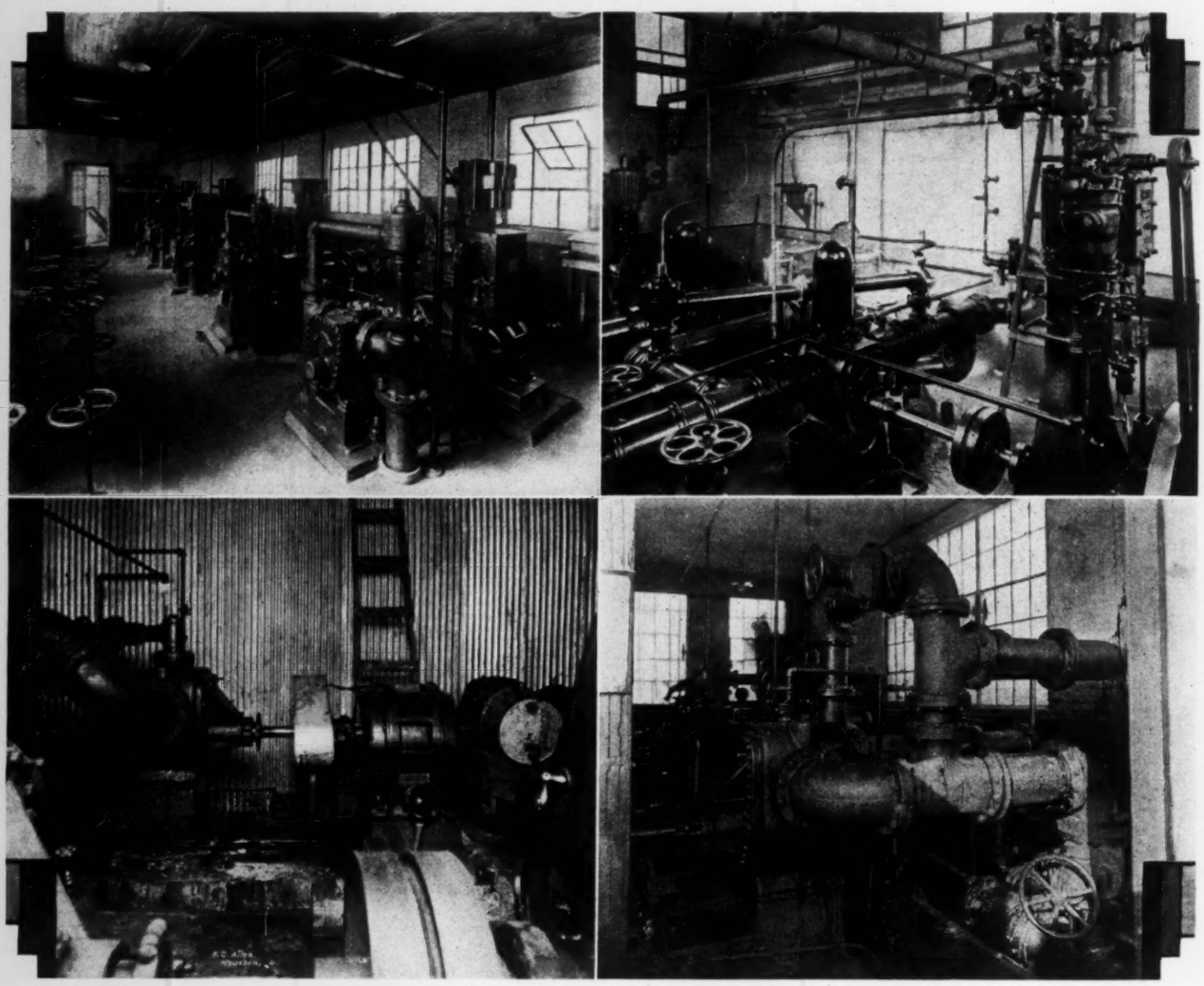


FIG. 6—PIPE LINE WARMED BY EXTERNAL ELECTRIC HEATER



FIGS. 7, 8, 9 & 10—SEVERAL PUMPING AND PIPING INSTALLATIONS WHICH HANDLE HEAVY, VISCOUS LIQUIDS

viscosity which takes place in characteristic oils manufactured for this type of service.

The physical characteristics of oil No. 1 are:

Specific gravity	0.925
Pour point	50 F
Flash point	610 F
Fire point	700 F

It will be noted that at 100 F, the viscosity of this oil is above 20,000 seconds (Saybolt Universal scale) whereas at an operating temperature of 250 F, this drops to less than 200 or only one-hundredth as much. As, usually, the operating temperature is well above 250 F, it is obvious that the pumping problem when starting up the system cold is very different from that of maintaining the circulation at operating temperatures.

Great care is necessary in designing the heating and piping system to introduce as little friction as possible and to provide suitable means for pre-heating at least part of the oil to a point where it can be readily handled by the pump. It is particularly important to heat the oil at the suction of the pump, as a pump suitable for this service will work against considerable pressure if its suction is provided with sufficiently fluid material.

As oils are broken down by heat, it is important that the heating be done under conditions most favorable to the prevention of this action, which is known as "cracking" in oil refinery parlance. Oil is a very complex mixture of hydro-carbons having different volatile points. Oil of the kind described has already been carefully distilled and re-distilled to eliminate all possible low temperature volatiles and it can be safely heated to high temperatures in a closed system if all necessary precautions are observed.

Importance of Definitely-Directed, Uniform Flow

It is essential that large heating surfaces of low temperature gradient be used and that a rapid, definitely-directed and uniform flow of the oil be maintained over all parts of these surfaces. There is always a film on the heating surface and the thinner this film, the lower the temperature gradient will be for a given energy transfer. An important function of the thickness of this film is the temperature, velocity, and pressure at which the oil is circulated over the surface.

Thermal Storage of Heater

Another problem of importance is the thermal storage capacity of the heater; the lower this heat storage capacity, the less rise will occur when it is necessary to interrupt or reduce the circulation. An ideal heater would be one having no heat storage capacity in itself. This is, of course, impossible to obtain. However, the more

nearly this can be approached, the better the operating results will be and the more accurately the operating temperature can be maintained under conditions of variable heat demand.

A Typical Job

Fig. 11 shows a typical installation in a large chemical plant. The equipment consists of a closed, jacketed pressure vessel in which compounds are heated; an arrangement of piping; two heating units; a motor driven circulating pump; an expansion tank, and all the necessary pressure and temperature regulating and control apparatus.

For this particular system, electricity was chosen as the heating medium because of its low stored heat capacity; accuracy of automatic temperature control; and the flexibility of manual control to suit the different operating conditions.

It is important that the oil connections to the kettle jacket be properly designed to secure uniform heat flow throughout. This becomes increasingly important for larger kettles. The hot oil inlet should be at the bottom with a number of top outlets connected to the return pipe leading to the pump.

Wherever possible, valves should be avoided in the circulation lines except where it is advisable to have several kettles in multiple and to alternate in heating different kettles. Where valves are present, care must be exercised to avoid closing them while the pump is in operation, although damage due to this possibility is safeguarded against by a maximum pressure switch which will stop the motor and the heat input upon any undue rise of pressure, and by a pressure relief valve connected from the discharge side of the pump back to the return which is also connected through the expansion tank to atmosphere, so that no excess pressure due either to pumping or overheating, can occur within the system.

For pipes over 3 in. in diameter, special flanged and welded fittings with suitable gaskets were used, but for pipe up to 3-in., special high test, screwed fittings were used. Fittings are available for service up to 300 lb. working pressure at 750 F for hot oil or vapor. Good threads with metal to metal fit are usually essential. Great care must also be exercised in laying out the piping with adequate provision for expansion.

Selecting the Pump

In the selection of pumps, it is important to select a type which will satisfactorily handle oil at the maximum temperature. A pump which would be satisfactory for cold oil may bind when exposed to extreme temperatures. It is also essential that the pump packing be suitable for high temperature and that the pump be capable of handling air and vapor as well as fluid, so that there is no danger of its becoming air bound. A positive action pump is necessary in order to force the fluid through the pipes when starting and it is essential that ample horsepower be provided to take care of the overload when starting up.

Filling the System with Oil

After the system is properly installed and tested, it should be filled with the oil selected. This oil is supplied from the refinery in metal drums and must be kept sealed to avoid being contaminated by moisture before it is needed. In order to fill the system properly, the oil should be pre-heated, preferably by setting the steel drums in a tank of hot water of approximately 200 F, for several hours so that the oil may become

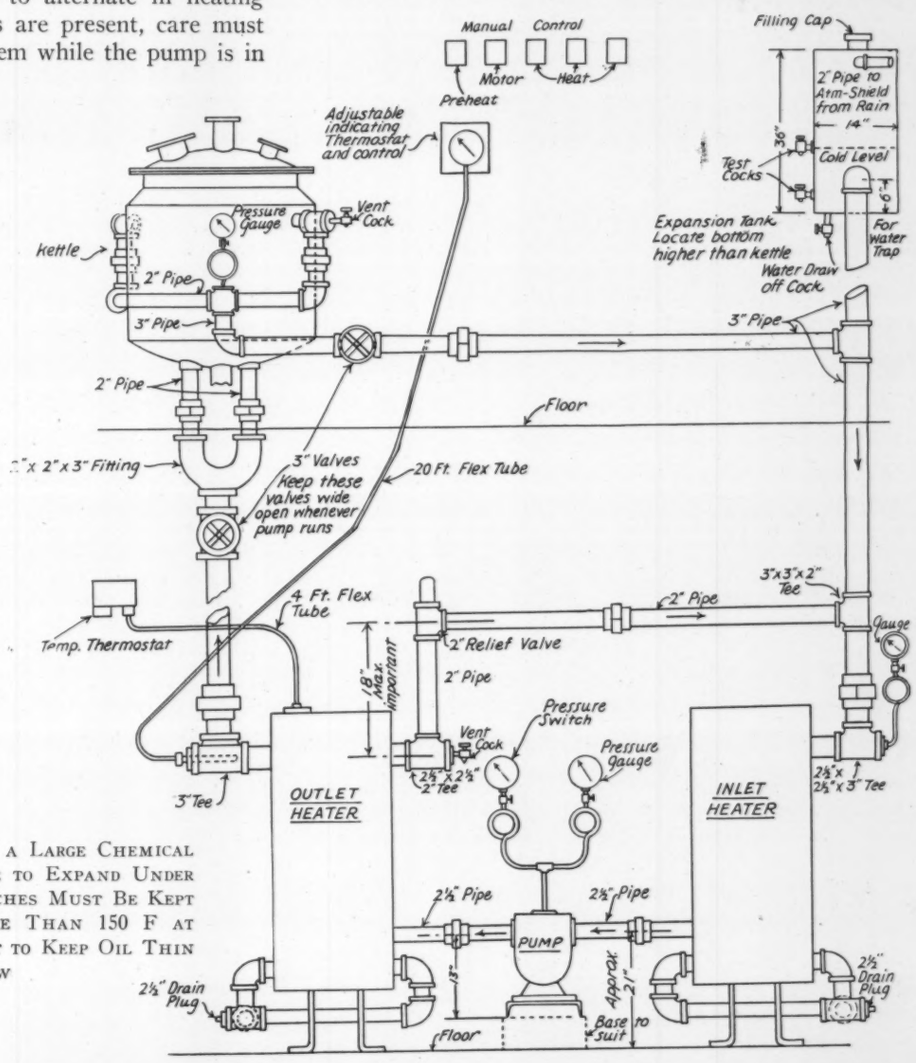


Fig. 11—Typical Installation in a Large Chemical Plant. The Pipe Must Be Free to Expand Under Heat. Gages and Pressure Switches Must Be Kept Away from Heat to Avoid More Than 150 F at Gage, but Near Enough for Heat to Keep Oil Thin Enough to Flow

thoroughly warm throughout. At this temperature, it is easily introduced into the system, preferably through the filling cap at the expansion drum.

During the filling period, the pump should be running to help pull the oil down into the piping. Vent cocks should be provided wherever air is likely to be trapped in the system and these should be opened as necessary

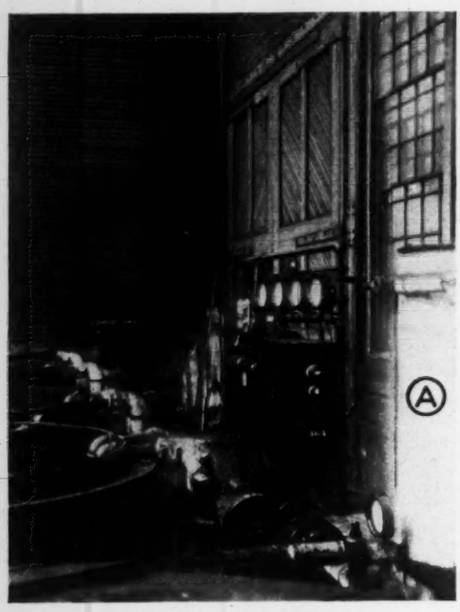


FIG. 12—HOT OIL SYSTEM FOR DRAWING THE TEMPER OF STEEL MILL ROLLS. THE OIL IS HEATED IN THE SPECIAL HEATER *A* AND PUMPED TO THE TREATING VATS

to eliminate it. In some cases, special air strainers can be used for this purpose.

Starting Up the System

After the system is completely filled with oil up to the normal cold level in the expansion tank, which is readily determined by the test cocks on the tank, it is ready for starting. If it is to be started up immediately while the oil is still warm, no additional pre-heating is required, it only being necessary to throw on as many stages of heat as are desired and set the automatic control for the proper temperature.

However, when starting up the equipment after it has cooled down, approaching normal room temperatures, it is necessary to pre-heat the main body of oil before the pump can be started. This is done by throwing a special pre-heating switch, and as soon as the temperature has been raised to approximately 200 F, this switch is thrown to the running position which starts up the pump and normal operation then proceeds.

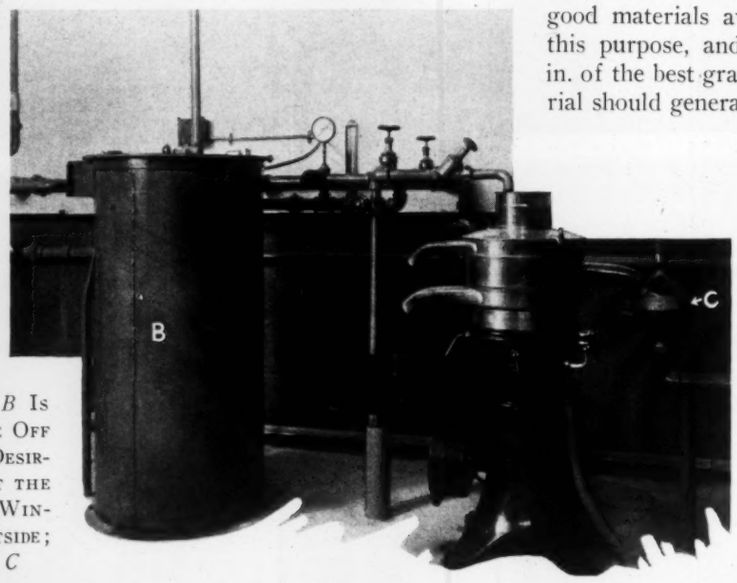


FIG. 13—OIL PURIFICATION SYSTEM; *B* IS THE HEATER. AS HOT OIL MAY GIVE OFF INFLAMMABLE VAPORS, IT IS OFTEN DESIRABLE TO PROVIDE A CLOSED FITTING AT THE DISCHARGE NOZZLE, WITH GLASS PORT WINDOW AND A VENT LEADING TO THE OUTSIDE; THIS ARRANGEMENT IS SHOWN AT *C*

Preventing Contamination of the Oil

A system of this kind properly installed and properly filled with a suitable fluid can be operated for long periods of time without any serious deterioration of the fluid. However, it is essential that the heating chambers be designed for inspection or cleaning if necessary. It is also very important to avoid any possible contamination of the oil by moisture, and the expansion tank must be located indoors and care be taken in arranging the piping to atmosphere so that no condensation can drain back into the tank. Even a few drops of water mixed with the oil will cause very serious foaming which is difficult to eliminate. Care must be taken to trap any possible moisture in the bottom of the expansion tank where it can be drained out without getting into the system.

Samples of the oil should also be drawn occasionally from the system at one of the lower vent cocks and tested to determine whether it is retaining its proper physical characteristics. As necessary, some new oil can be added to the system at the filling cap of the expansion tank.

Arrangement of Control Equipment

In mounting gages and pressure switches on a hot oil system of this kind, it is necessary to use care in locating them far enough away from the hot oil circuit to avoid damage to the instruments, as they usually should not be subjected to a temperature above 150 F. It is not satisfactory to mount them so far away from the pipe that the fluid will entirely congeal. If this occurs, satisfactory operation will not be possible.

It is also necessary to mount the relief valve not more than 18 in. from the hot oil line to which it is connected. Otherwise it may be so sluggish, due to the congealed fluid in it, that it will not function properly.

Insulating

For economical operation, it is important that care be used in applying heat insulation to all parts of the system. There are a number of good materials available for this purpose, and at least 3 in. of the best grade of material should generally be used.

The New Hotel Waldorf-Astoria —A Description of the Ventilating and Air Conditioning Systems

By the Staff of Clyde R. Place*

(*Edited by Ernest Williams*)

The particularly interesting features of the ventilating and air conditioning systems in New York's new Hotel Waldorf-Astoria are described by the staff of the consulting engineer, no attempt being made to discuss the systems in too much detail.

About twelve per cent of the ventilating load in the building is conditioned completely; the system was designed for the maximum possible in flexibility and operating economy.

THE ventilating system for the new Hotel Waldorf-Astoria (which was described last month) is most complete, and designed for the economic accomplishment of the desired results.

All the rooms in the lower section of the building intended for human occupancy or congregation are provided with complete supply and exhaust systems; every toilet room and bath room, regardless of location, is ventilated.

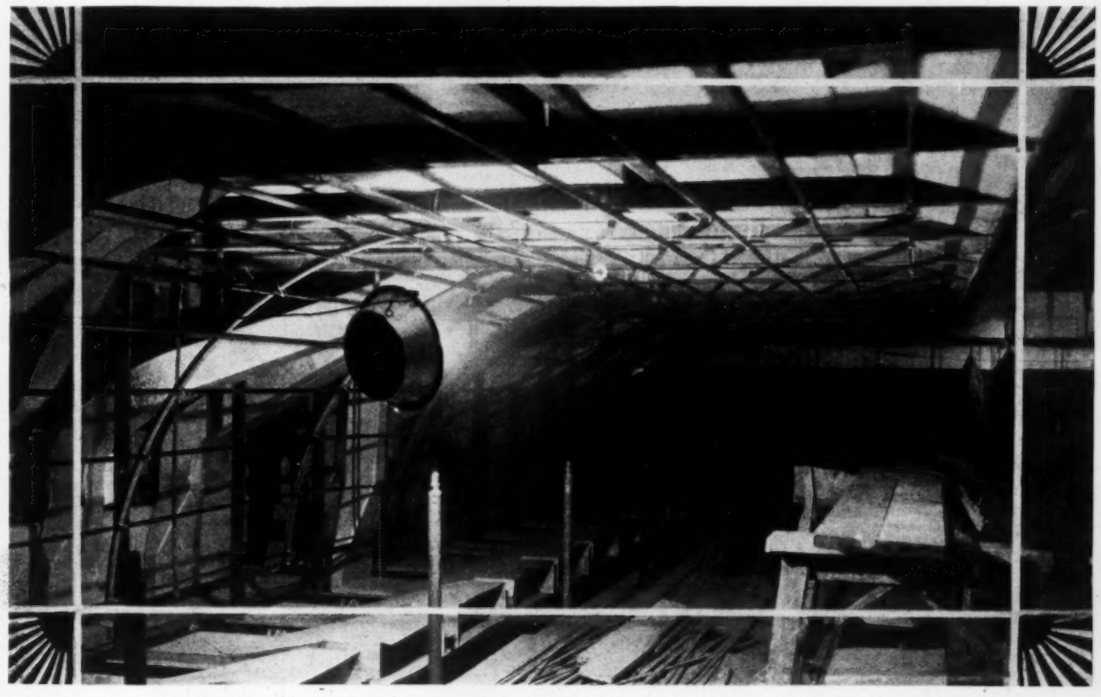
A Million C.f.m.

Twenty supply fans and thirty-five exhaust fans, exclusive of the conditioned air equipment, are necessary to provide ventilation. These fans handle a total of over a million cubic feet of air a minute; the equivalent of a complete change of air for the whole building in twenty minutes! A special signal system has been designed to aid in the ventilating of the entire structure.

The supply ventilating units are, except in one or two cases, of the draw-through type; that is, the air is drawn through the filter or heater and washer, then the heaters or reheaters, and discharged directly into the duct system from the fan.

For a job of such proportions the supply ducts are amazingly short, due in part to locating the fan units adjacent to the rooms on which they operate. Another reason is that the rooms ventilated are so large that most of them have separate supply and exhaust units. It is expected this layout will provide excellent operating conditions, economical in operation due to low resistances.

*Consulting engineer, New York City.



AIR CONDITIONING DUCTWORK FOR THE MAIN BALL ROOM. THE LOWER ROW OF OUTLETS CAN BE SEEN AT THE LEFT, AND THE UPPER ROW OF OUTLETS AT THE EXTREME RIGHT. THE STEEL CATWALK MAY BE SEEN AT THE EXTREME LEFT

The grouping of the exhaust units differs greatly from the supply, mainly because of structural conditions. The fans are located generally in the upper section of the building and are grouped together into four main fan rooms. Two rooms are at the main offset floor, one in either wing, and the other two above the main roof, one in each tower.

The Ducts

Four main shafts house the individual ducts to the fans in the various rooms, two of which go to the lower fan rooms and the other two go through the tower section to the rooms above the main roof. The aggregate area of these shafts is about 525 square feet. This figure may appear to be smaller than expected, but it must be borne in mind that *some* of the exhaust fans are located on the lower floors for economy. They are those units which could not possibly have any objectionable odors to discharge near windows.

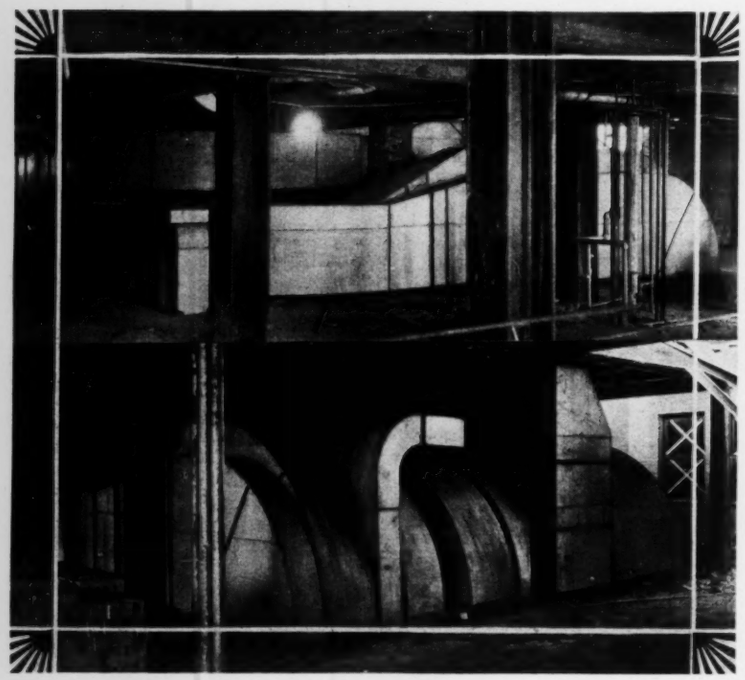
The largest single duct installed has an area of 50 square feet.

All toilet rooms, whether exterior or interior, are ventilated with separate duct systems, the main toilets throughout the lower section being on separate fans from the bedroom toilets. There is nothing out of the ordinary about the bedroom toilet exhaust system except that sheet metal ducts are used throughout and grouped in vertical sets for balancing of resistance and ease of maintaining the desired quantities at each inlet.

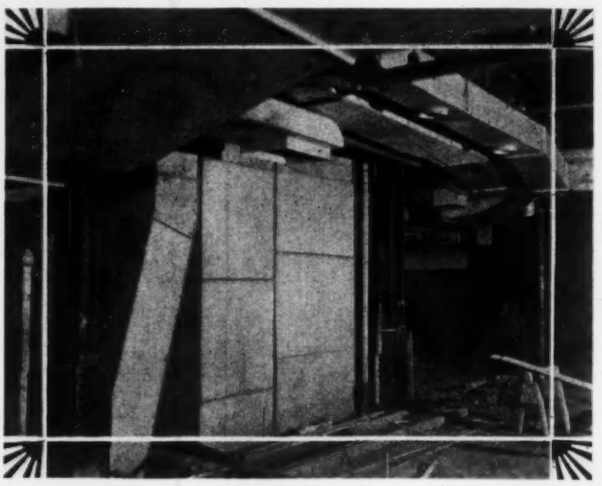
120,000 C.f.m. for Main Kitchen

One room which deserves special mention is the main kitchen. The supply and exhaust air for this one room is taken care of by six fan units, handling a total of 120,000 cubic feet of air per minute.

View of the Fourth Floor Vent Shaft Offset



Main Vent Shaft Offset at the Fourth Floor Level. Note the Streamline Bends and Angle-iron Bracing



View of Main Vent Shaft at Basement Level. Ducts for Laundry Ventilation Are at the Right

Twelve Per Cent of Ventilating Load Conditioned

The air conditioning system for the building supplies over 120,000 c.f.m., amounting to about 12 per cent of the total ventilating load, and is handled by ten units. The units are designed to meet the individual requirements of each room, for flexibility and for economy in operation.

The chilled water to, and return water from, these units is conducted in two main risers and is circulated through the cooling coils in the refrigerating compressor which is located in the basement. This compressor is used for air conditioning only, and is in no way connected with the refrigerating plant for the building.

Condensing Water Used in Plumbing System

An 8-in. C. W. line runs from the distributing header to the air conditioning apparatus. The air conditioning equipment, when operating at 100 per cent capacity, will use approximately 750 g.p.m. of water for condensing purposes. This quantity of water closely approximates the demand of the building for the fixture supply.

Since the condensers using this water are operating at pressures which are far below that of the cooling water, it is feasible to use the condensing water in the plumbing system.

The water from the condenser is taken by a somewhat unusual piping arrangement. The suction tank is equipped with two sets of ball cocks, one set approximately two feet below the other. The upper ball cocks are connected to the discharge from the condenser and the lower cocks are connected to the 8-in. line from the service header. A line from the condenser line is carried up and discharges behind a leader trap.

The operation of this arrangement is as follows:—When the condensers are in operation the condenser water flows into the suction tank. If the house demands do not exceed the condenser discharge the tank level rises and closes the upper ball cocks. The water then rises in the pipe connected to the leader trap and dis-

charges to the sewer. If the house demand exceeds the condenser discharge the level of the water in the tank falls and the ball cocks connected to the header open and augment the supply. The system is entirely automatic; the ball cocks are duplicated and separately valved so that a possible failure will not cripple the system.

The Conditioning Units

The air conditioning units consist of the usual dehumidifying washers, heaters, reheaters, controlling dampers and fans; the supply to conditioned spaces, and return ducts where placed outside the conditioned spaces, are covered for economy of operation.

The rooms cooled include the main dining rooms, main ball room and its foyer and relating spaces, the barber shop and the beauty parlor. The ventilating system for the main lobby has been so designed that the air may be cooled when the refrigeration is not demanded for some of the other rooms mentioned above.

System in Ball Room a Problem

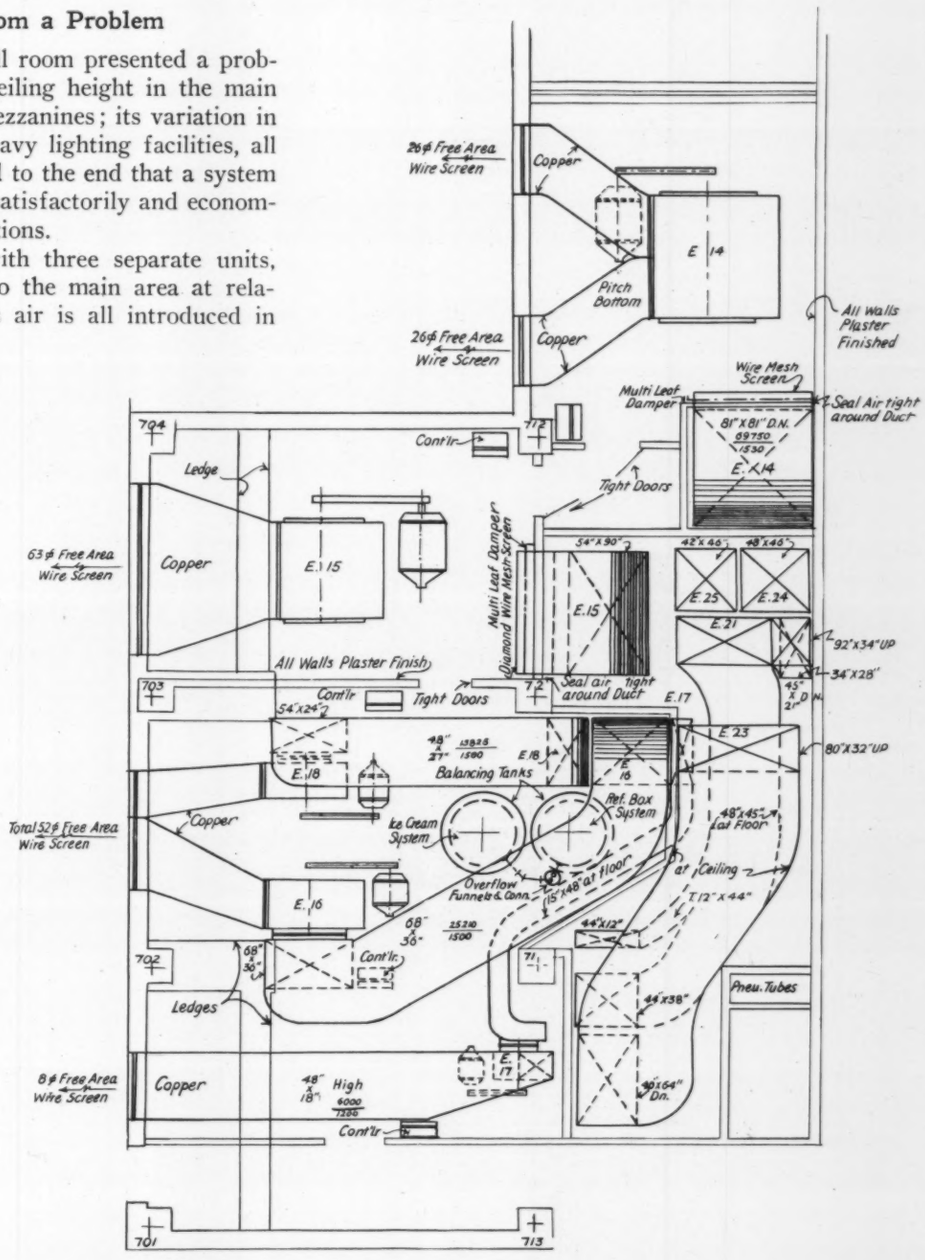
The system for the main ball room presented a problem of design, with its high ceiling height in the main area and low ceilings on the mezzanines; its variation in occupancy and uses, and its heavy lighting facilities, all of which were carefully studied to the end that a system might be designed to function satisfactorily and economically under all probable conditions.

The result was a system with three separate units, two of which supply the air to the main area at relatively low temperatures. This air is all introduced in the high parts of the room. The other unit is designed to introduce the air around the boxes and in the lower parts of the room and is intended for a lower temperature differential than the air introduced at the higher points.

Removing Heat from Lights

The heavy cove lighting in the high ceiling presented difficulties. It was apparent that if the cooling system were designed to take care of all this lighting it would have entirely too large a capacity for normal use. If the system were designed by disregarding this lighting, conditions would frequently occur when the system was inadequate. The problem was eventually solved by the use of an exhaust along the back of the cove to remove the heat when the heavy lighting loads were on, without taxing the cooling plant. This unit also serves as the flushing or smoke exhaust.

The salient features in the design of the heating, ventilating and air conditioning equipment for this structure were that each system was to be made as complete and economical in operation as possible; flexibility and operating economy in a hotel of this size are extremely important factors.



PART PLAN OF THE 20TH FLOOR FAN ROOM. (DRAWN FROM THE CONSULTING ENGINEER'S BLUE PRINTS)

Proper Care of Brine Aids Prevention of Corrosion

By R. C. Doremus*

PROBABLY the greatest enemy of our machine age is the corrosion of metals and retardation of corrosion is, therefore, a problem of great economic significance.

To the operator of every refrigerating plant, whether it is for air conditioning, the manufacture of ice or ice cream, cold storage, meat packing, dairy foods or any one of a hundred other enterprises, the corrosion of metals, particularly metallic piping used in the handling of brines, is a question of major importance. The plant engineer formerly paid little attention to his brine except for an occasional test of its strength to assure that it contained a sufficient quantity of salt to prevent freezing under normal operation. A salometer for salt brine, or a hydrometer for calcium brine, with a reputable table of strengths such as that issued by the *A. S. R. E., N. A. P. R. E.*, the *Bureau of Standards,* or the chemical manufacturers, told him his brine gravity and corresponding freezing temperature. Beyond this, his knowledge of brines and their characteristics was, on the average, dangerously brief.

Maintenance of a refrigerating plant for air cooling or any other use includes the proper care of the brine used; the piping must be protected from the corrosive action of brine, as well as must the other equipment.

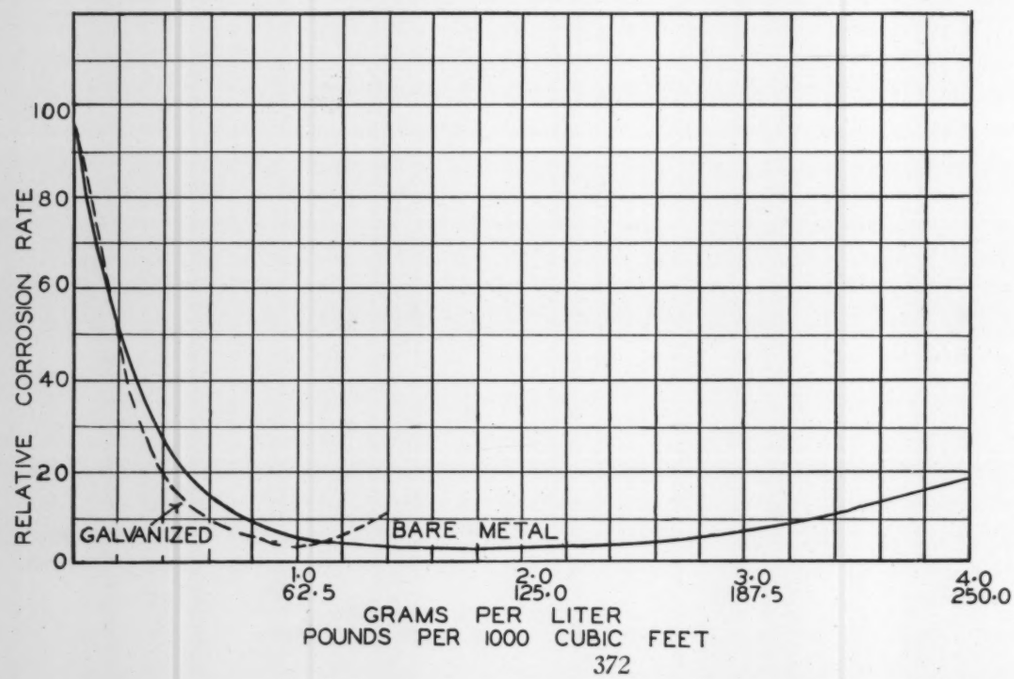
During the last five years, much has been learned about brine corrosion, largely through the efforts of the Bureau of Standards, the A. S. R. E. and the N. A. P. R. E. We asked Mr. Doremus to review this fund of information in a brief, practical article for the engineer who maintains a refrigerating plant and refrigeration piping; his article will be found a handy reference.

During the last five years there has been an increased interest in the kindred refrigerating industries as to the causes and effects of brine corrosion. Reports of research work done by the *A. S. R. E.* Corrosion Committee, various technical schools and by the chemical manufacturers of the chlorides used in making brine, has considerably fortified our knowledge of the subject. Valuable studies have been made in laboratories, and confirmed in actual plant operation, of such functions of corrosion as strength, electrolysis, acidity and alkalinity, and proper emphasis placed on the importance of maintaining a correct *pH* value in the brine, as well as other means of retarding corrosion. Much of the technical information submitted herein has been obtained from the above mentioned authoritative sources. The curves showing rates of corrosion and effects of retarders have been contributed by Messrs. R. P. Russell, J. K. Roberts and E. L. Chappell from tests run by them at the *Massachusetts Institute of Technology.*

Strength

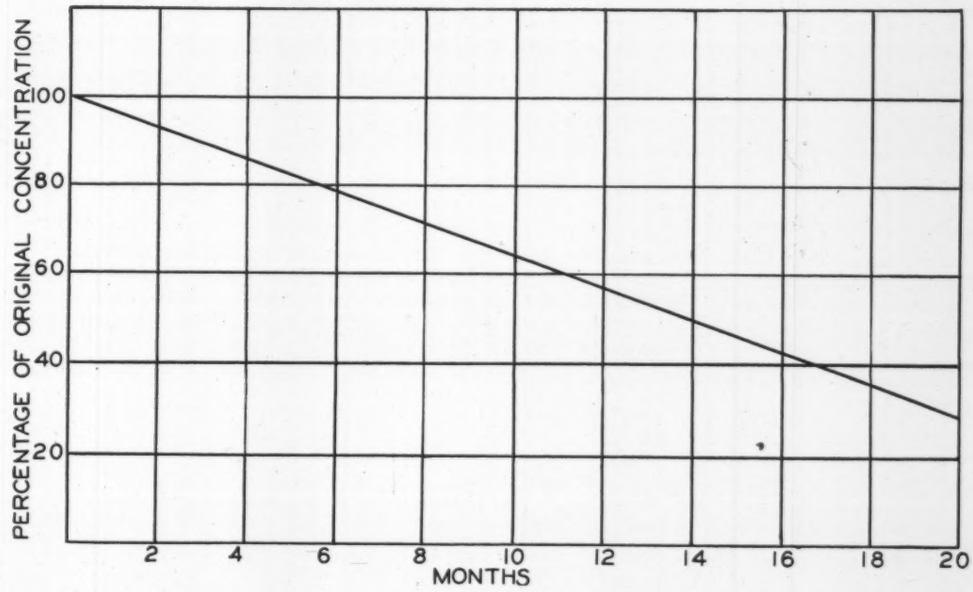
Corrosion is more active in weak brine than it is in

* George B. Bright Co., Refrigerating Engineers and Architects, Detroit, Mich.



CORROSION OF BARE STEEL AND GALVANIZED STEEL IN SODIUM CHLORIDE BRINE TREATED WITH DISODIUM PHOSPHATE

Note—Disodium Phosphate Gives 95 Per Cent Protection with a Charge of 70 Lb. Per 1000 Cu. Ft. Brine. If Metal Is Bare, 100 Lb. Charge Is Recommended.



DECREASE IN CONCENTRATION OF SODIUM DICHROMATE IN CALCIUM CHLORIDE BRINE

strong brine, probably due to the fact that the stronger the brine the less oxygen there is present in the solution. For example, in calcium chloride brine having a specific gravity of 1.05, the amount of corrosion on steel is twice the corrosion in a brine having a specific gravity of 1.18. Likewise, in sodium chloride brine, the amount of corrosion for a brine having a specific gravity of 1.05 is three times that of a brine having a specific gravity of 1.18. From this it is evident that the amount of corrosion of steel in either kind of brine is minimized when the brine has a strong specific gravity, and the amount of corrosion increases as the brine grows weaker. This is shown by the chart on page 374.

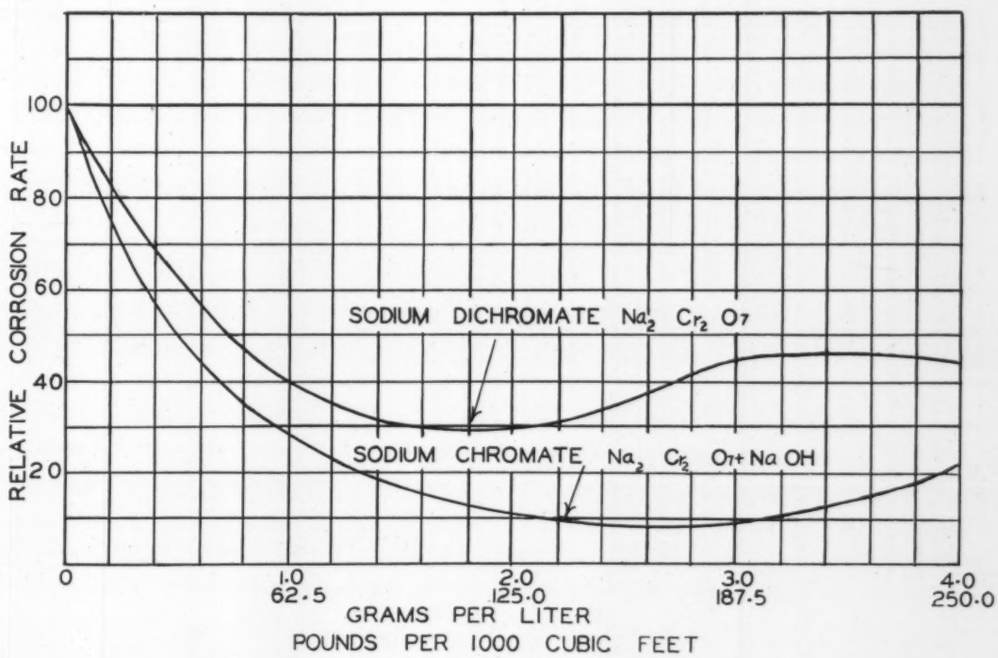
Electrolysis

Corrosion by electrolysis is caused when two dissimilar metals are in contact or in close proximity and immersed in a brine solution, whether or not there is any known electric current passing through the solution or piping. The electric current is set up inasmuch as the two metals form a galvanic cell and the brine is the electrolyte. Bronze and iron are frequently used together in parts of the same brine piping system by error. In such cases, it has been found that the iron forms the anode and the bronze forms the cathode of this elementary galvanic cell. Particles of iron leave the metal and go into solution in the brine and eventually find their way to the bronze parts, where the electric charge is deposited and hydrogen gas liberated; and the brine gradually becomes discolored a dark reddish brown color by the accumulation of iron oxide. Eventually, the iron is badly corroded or graphited.

A way to avoid electrolysis is to make all component parts of the system of one metal. Care must also be taken not to pass an electric current through the brine, as would happen if a motor were grounded to a brine pipe.

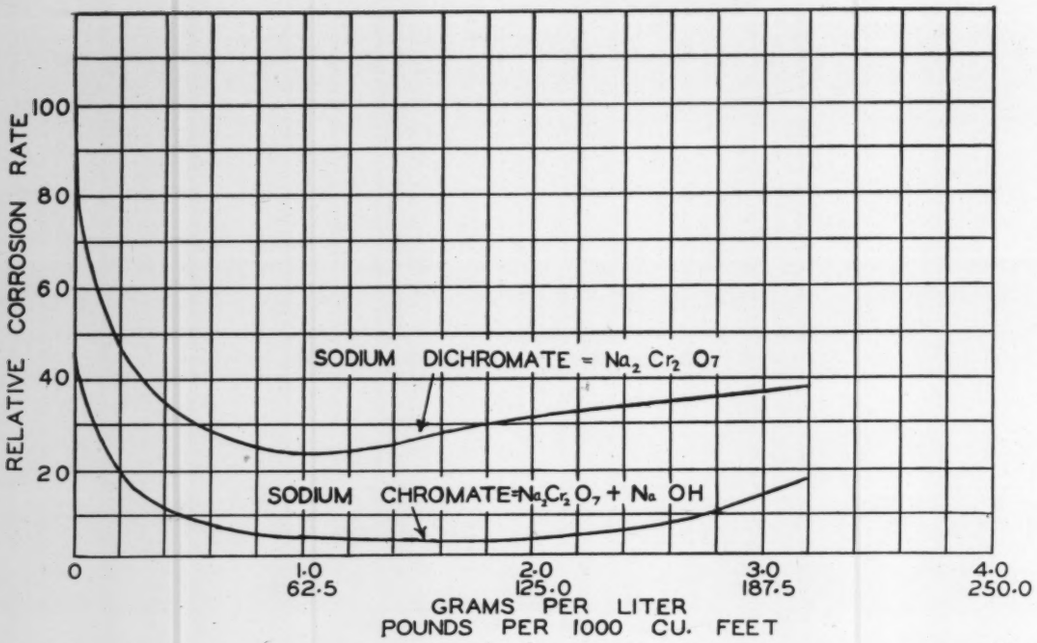
Acidity

It is well known that no matter what kind of brine



CORROSION OF IRON AND STEEL IN SODIUM CHLORIDE BRINE TREATED WITH SODIUM CHROMATE OR SODIUM DICHROMATE

Note — Sodium Chromate Gives 90 Per Cent Protection with a Charge of 150 Lb. Per 1000 Cu. Ft. Brine. 200 Lb. Is Recommended to Take Care of Loss During First 6 or 8 Months.



CORROSION OF IRON AND STEEL IN CALCIUM CHLORIDE BRINE TREATED WITH SODIUM CHROMATE OR SODIUM DICHROMATE

Note — Sodium Chromate Gives 95 Per Cent Protection with a Charge of 70 Lb. Per 1000 Cu. Ft. Brine. 100 Lb. Is Recommended to Take Care of Loss During First 6 or 8 Months.

may be used, it should not become acid or the acidity will attack the metals comprising the brine piping system. Any open system will gradually become acid by absorption of carbon dioxide from the air, and the piping system should be arranged for a minimum of splash, or leaks which bring air in contact with brine, for the same reason.

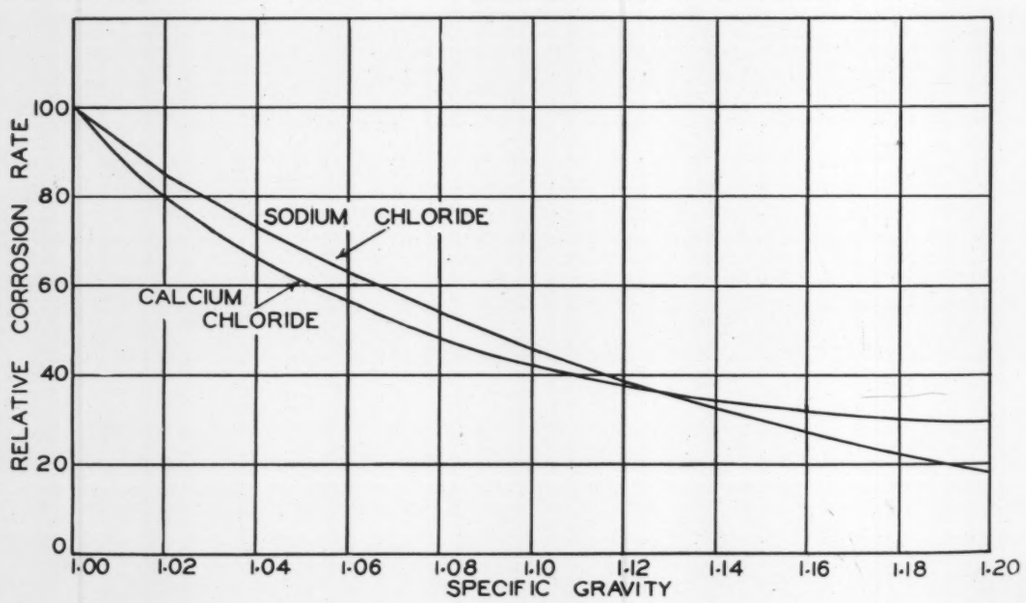
The chemical reaction that takes place is: $2\ Ca\ Cl + H_2O + CO_2 = 2\ H\ Cl + Ca_2CO_3$. This means that two molecules of calcium chloride and one molecule of water and one molecule of carbon dioxide absorbed from the air produces two molecules of hydrochloric acid and one molecule of calcium carbonate. In an ice tank, for instance, carbon dioxide is continually absorbed by the brine, and the brine, therefore, continually becomes more acid, with its detrimental effect on corrosion of all of the iron and zinc parts exposed to the brine. Therefore, the acid should be neutralized with caustic soda or lime to a point where the brine is slightly alkaline. Alkalinity protects iron and steel from corrosion.

Alkalinity

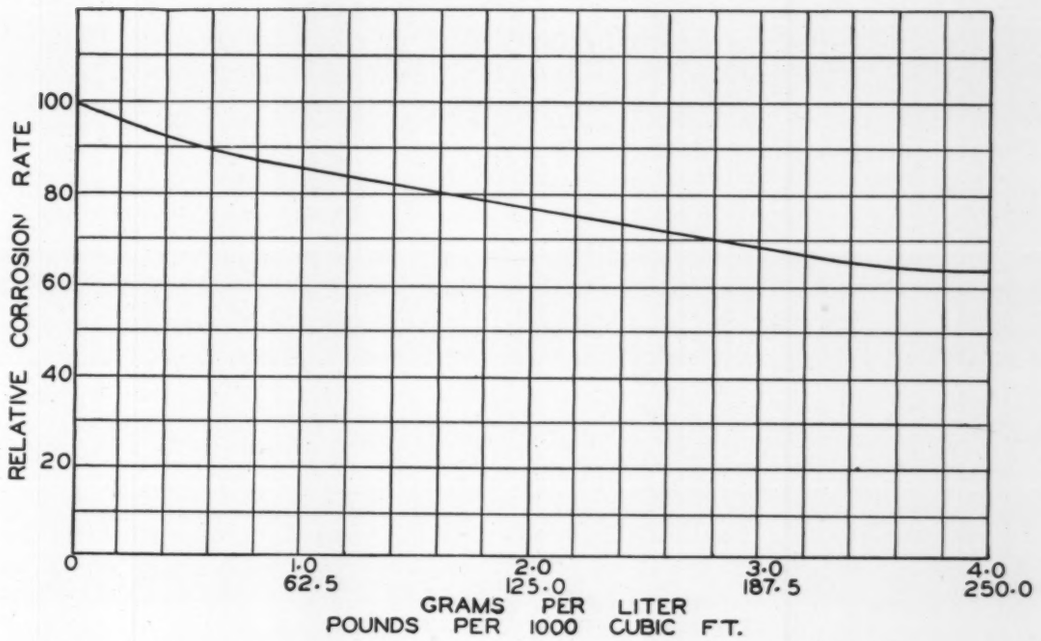
High alkalinity in brine has practically no noticeable effect on iron and steel, but does have a corrosive effect on brass and zinc, especially in sodium brines. Therefore, the brine should not be too alkaline. For example, high alkalinity will attack zinc and form zinc hydroxide, which is a gray soft soap, and is frequently called "zinc soap." The galvanizing will come off from ice cans in spots if the alkalinity is slightly strong and will be removed in sheets and strips if it is extremely strong. This is frequently noticed when making new brine from calcium chloride that may be initially high in alkalinity, or from a bad ammonia leak into the brine solution. Experiments have shown a minimum of corrosion to all metals in the system, whether it be an open or closed system and no matter what chloride is used in the brine, if the brine is slightly on the alkaline side of the neutral point.

Tests

Whether the brine be new or old, it should be fre-



CORROSION OF STEEL PIPE IN CALCIUM AND SODIUM BRINE OF DIFFERENT GRAVITIES



CORROSION OF IRON AND STEEL IN SODIUM OR CALCIUM BRINES TREATED WITH SODIUM SILICATE (WATER GLASS)

Note—Sodium Silicate Gives 30 Per Cent Protection with a Charge of 200 Lb. Per 1000 Cu. Ft. Brine.

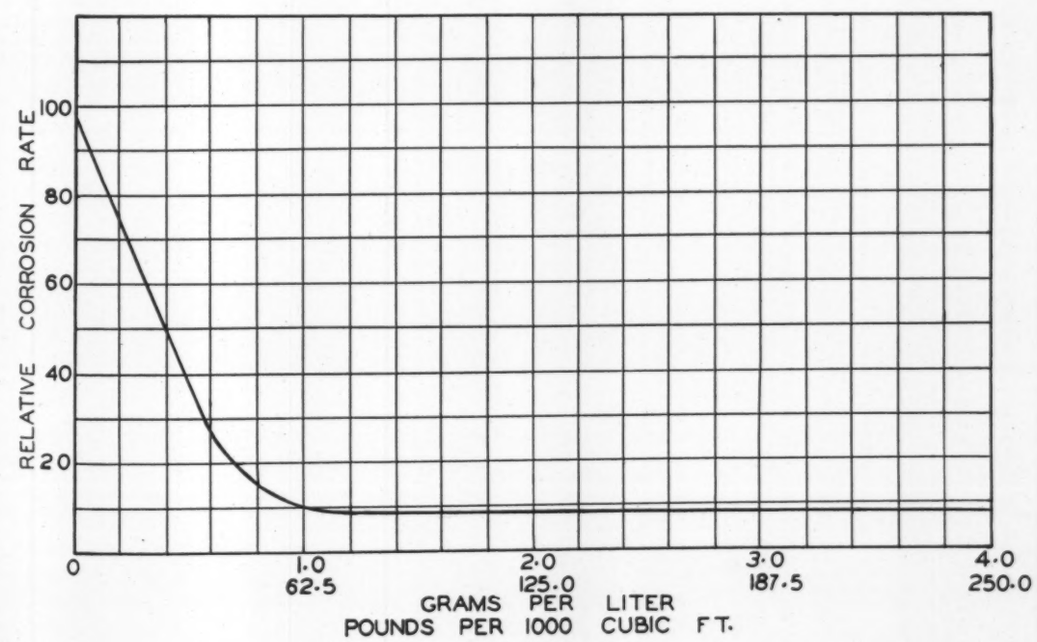
quently tested by the engineer, not only for proper strength with a hydrometer, but also for acidity or alkalinity. Most engineers are familiar with the litmus paper test. If the brine is alkaline it turns red litmus "blue"; if the brine is acid it turns blue litmus "red." In either case, a slight shade of red or blue indicates a relatively small amount, whereas a deep shade indicates an extreme amount.

However, litmus indicates only whether it is acid or alkaline and the relative amount of acidity or alkalinity must be determined by some other means. Inasmuch as the degree of corrosion seems to depend upon the hydrogen ionization or oxygen content of the brine, a convenient scale has been adopted whereby any sample of brine may be tested and given a *pH* value. This scale of values is arranged in order to indicate relatively the amount of acidity or alkalinity in the brine. In this scale, 7.0 is neutral. Any *pH* value below 7.0, such as 6.8, 6.4, 6.0, etc., indicates an acid condition. Any value of *pH* in excess of 7.0, such as 7.5, 8.0, 9.0, 10.0, would indicate an alkaline condition.

There are various indicators used as a scale for *pH* values and the use of these indicators will change the color of the brine sample to various shades, which will indicate the *pH* value. One brine testing outfit gives the following shades of colors for various *pH* values:

Color	*pH Value*
Red	up to 4.0
Orange	5.5
Yellow	6.5
Greenish Yellow	7.0
Green	8.0
Blue	9.5
Violet	above 10.0

Bromthymol Blue is sometimes used as an indicator, and this will show "yellow" in acid and "blue" in alkaline brines. The table for use of this indicator is as follows:



CORROSION OF IRON AND STEEL IN SODIUM CHLORIDE BRINE TREATED WITH ZINC DUST

Note—Zinc Gives a 90 Per Cent Protection with a Charge of 60 Lb. Per 1000 Cu. Ft. Brine.

Color	*pH* Value
Yellow	6.0
Green	7.0
Blue	7.6

Thymol Blue is sometimes used with brines of higher alkalinity; the test table of *pH* values is as follows:

Color	*pH* Value
Yellow	8.0
Green	8.8
Blue	9.6

Phenol Red is also used as an indicator, and its table of *pH* values is:

Color	*pH* Value
Yellow	6.8
Orange	7.6
Red	8.4

Cresole Red is also used as an indicator:

Color	*pH* Value
Yellow	7.2
Orange	8.0
Red	8.8

Methyl Orange, Methyl Red and several other indicators are frequently used, but it would be confusing to list all of them and their shades of orange and red are not easily differentiated except by an experienced chemist.

Use of Phenolphthalein

Phenolphthalein is popularly used inasmuch as it is the alkaline indicator that is usually found in plants where water softening equipment is used. It is simply necessary to measure a 100 cc. graduate of brine from the system and add about three drops of "P" indicator:

	Color	*pH* Value
Acid:	Colorless	below 7.0
Neutral:	Colorless	7.0
Alkaline:	Pink	above 7.0

If the sample being tesed is alkaline, it will have a pinkish color. If the solution has no pink color, it is either neutral or acid. If the brine is used in an ice tank where galvanized cans are used, it should have a *pH* value of 7.5 to 8.0. If there are no galvanized cans used nor any bronze agitator propellers, or if cans are used from which the galvanizing has completely disappeared, the alkalinity should be preferably higher in order to give better protection to iron and steel parts, and in this case the *pH* value should be 9.0 to 9.5.

For a *pH* value of 8.0, this 100 cc. sample solution should be titrated with eight drops of 1/10th strength, normal acid, and this amount of acid should neutralize the alkalinity in the sample and change the pink color to "colorless." As the acid usually used in a water softening chemical test set is 1/50th strength normal acid, the titration will require about five times as much as the 1/10th strength, or, in other words, forty drops to change the pink color to "colorless," provided the *pH* value is 8.0. From this it is readily seen that if the sample is higher in alkalinity than 8.0, it will require more acid to neutralize the alkalinity, and if the *pH* value is not as great as 8.0, it will require less acid. This, therefore, becomes an easy method of testing the brine, which can be readily made by any plant engineer using the testing chemicals and cabinet at hand.

If the brine is too alkaline, as will be indicated by an excessive amount of acid required to neutralize the sample, the corrosion to zinc, bronze, etc., will be excessive. In this case, the excessive alkalinity should be neutralized with acid to the proper point. This may be done by hydrochloric (muriatic) acid if the operator is experienced in handling acids; it requires extreme care to dilute it in a wooden pail and drop it slowly into the brine at a point of rapid agitation.

Another effective method is to dissolve CO_2 gas slowly into the brine at a point of rapid agitation. This gas is sold by different carbonic gas companies for soda fountain use, and is therefore easily procurable at a cost of approximately 8c per pound in any locality. If the titration shows that the sample is either neutral or acid, the acidity should be neutralized with some alkali such as caustic soda (NaOH) or lime (CaO). Usually, a very few pounds will be required per thousand cubic feet, and this may be placed in a cloth bag at a point of rapid agitation.

Regardless of whether the brine needs treatment with acid or alkali to properly condition it, frequent titration tests should be made every two or three hours while the treatment is in process so that it may be stopped when the brine reaches the proper condition of slight alkalinity.

Sodium Dichromate

Inasmuch as minimum corrosion of zinc and bronze requires low alkalinity, while minimum corrosion of iron and steel requires high alkalinity, the brine must be kept at low alkalinity in order to favor the galvanizing if any is present. This means that the iron and steel will corrode faster than it would if the alkalinity were at a higher point, and therefore, in some cases, retarders have been used to retard the corrosion of the iron and steel parts of the system.

Research work has indicated that sodium chromate forms a very good retarder for either calcium chloride or sodium chloride brines. Inasmuch as the normal chromate is not so easily obtained, it is customary to use sodium dichromate ($Na_2 Cr_2 O_7 . 2 H_2O$) which in itself is an acid forming chemical, and therefore must be neutralized with alkali to form normal chromate and maintain a neutral brine condition.

If the brine is just slightly alkaline, as it should be, it is customary to use 100 lb. of sodium dichromate and 35 lb. of caustic soda per 1000 cu. ft. of calcium chloride brine. If it is new brine, it is probably already extremely high in alkalinity, in which case less caustic will be needed to neutralize the dichromate. In some cases, the amount mentioned as 35 lb. per 1000 cu. ft., may be reduced to 25 lb. or less. If too much alkali is used at this point, it should be quickly neutralized with acid to prevent damage to galvanized parts. In the case of either calcium-magnesium or sodium chloride brines, these quantities should be doubled, i.e., 200 lb. of sodium dichromate and 70 lb. of caustic soda.

Dichromate gives a yellowish color to the brine, removes rust from surfaces of tank, cans or piping and retards the corrosion, as may be seen by the curves, reducing the normal corrosion from 100 per cent to 5 per cent of what it might otherwise be. Care must be exercised in handling the chemical or in coming in contact with dichromate treated brine inasmuch as it has a tanning action on the skin, and will produce what is commonly called "chrome itch." Dichromate gradually spends itself in counteracting corrosion and usually 50 per cent of the original charge should be added every six months.

Disodium Phosphate

In the case of sodium chloride brines, the use of disodium phosphate ($Na_2 HPO_4 . 12 H_2O$) is general in the ratio of 100 lb. per 1000 cu. ft. wherever dichromate is impracticable. This treatment also reduces the corrosion from 100 per cent to approximately 5 per cent. It removes rust from metal surfaces and is a deterrent against rapid corrosion. It has one disadvantage in the fact that the complete charge of disodium phosphate usually must be added to the brine once every month. This material cannot be used in calcium or calcium-magnesium brines, due to the fact that calcium and magnesium phosphates are very insoluble in alkaline solutions and would form an objectionable sludge in the lower portion of the brine.

Zinc Dust

Commercial zinc dust (Zn), which is composed of approximately 94 per cent metallic zinc, 5 per cent zinc oxide and 1 per cent impurities, has also been used to some extent as a corrosion retarder in both sodium and calcium brines. It has been found to give a 90 per cent protection against rapid corrosion when used in the ratio of 60 lb. per 1000 cu. ft. of brine. The zinc dust is simply spread on the surface of the brine in order to have as much remain in suspension as possible. Probably the biggest detriment to its use is that the material is slightly heavier than brine and unless agitation is fairly rapid, the zinc dust settles to the bottom of the tank instead of remaining in suspension.

Costs

Naturally, the question of the advisability of using corrosion retarders depends largely on a cost comparison. In some instances their use is very well justified, whereas in old systems that are already extensively corroded, their use might be questioned or even considered inadvisable on account of the small savings afforded.

In a typical market, the prices of these chemicals are approximately as follows:

Sodium dichromate	6½ to 7 c per lb.
Disodium phosphate	7 to 8 c per lb.
Muriatic acid	2¼c per lb.
Caustic soda	4¼c per lb.
Zinc dust	8 c per lb.
Carbon dioxide	8 c per lb.
Lime	1.3 c per lb.

It will usually be found that the total cost of labor and material for treating the brine in a 120-ton ice plant will be in the neighborhood of fifty to sixty dollars per year. This will serve to give a rough idea of the cost in other types of refrigerating plants.

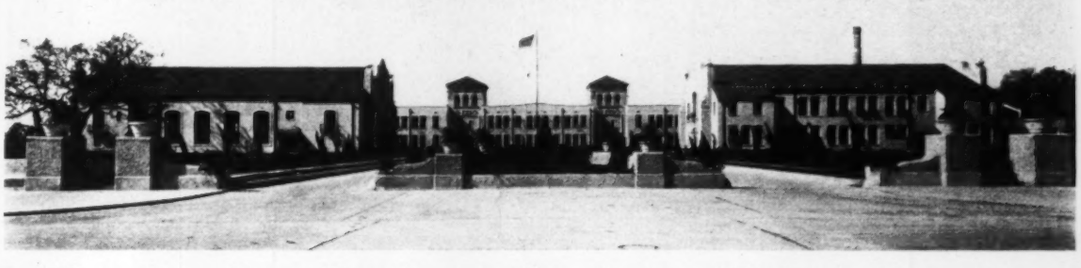
With the aid of such prices, it is not a difficult matter to make a cost comparison to determine whether or not the use of a corrosion retarder is advisable.

Conclusion

It will be seen that the use of corrosion retarders is not always mandatory, but should be determined on the basis of cost of treatment versus savings afforded.

It will be noted from the paragraphs on acidity and alkalinity content, that corrosion may be greatly retarded where zinc is present if the *pH* value is maintained at 7.5 to 8.0. If no zinc is present, and the problem is simply to protect iron and steel, the *pH* value should be from 9.0 to 9.5. In the case of all brines and all metals exposed to their corrosive action, the brine should be slightly alkaline with the *pH* value remaining between 7.5 and 9.5, according to the metals exposed to corrosion. Whereas the gravity of most brines is determined by the freezing point of the brine solution, the heavier gravity brines have a less corrosive effect on metals than the lighter gravities. Practically no cost is involved in frequent testing of brine condition for acidity or alkalinity, especially if testing equipment is at hand, and the proper treatment of brines will many times repay its cost in corrosion prevention.

Continuity of service is required in the operation of cold storage plants or air cooling or conditioning systems using brine refrigeration, for only brief shutdowns can be permitted for maintenance or repair. The vital necessity of observing the condition of the brine, brine piping, fittings, pumps and all parts of such piping systems hardly needs to be stressed. Regardless of the nature of the industry under consideration, the economic advantages of properly maintaining the brine, tanks and piping to retard the ravages of corrosion cannot be too strongly emphasized.



THE MILLION DOLLAR MURPHY HIGH SCHOOL IN MOBILE, ALABAMA. J. W. HULL OF MEMPHIS WAS THE HEATING AND PIPING CONTRACTOR

Operation and Maintenance of Air Filters

By W. G. Frank*

PERFORMANCE of an air filter depends not only on the design of the filter itself, the form of the filter screens, the structural and mechanical features, but rests in a large measure on correct installation, suitable operating conditions and maintenance.

We will assume that the type of filter best suited for the particular dust problem has been selected, that it has been installed so that the air flow is evenly distributed over the entire filter area, that the size is ample for the air volume to be handled and that the filter is readily accessible and the clean air connections are air-tight. What attention should be given to the filter for it to function without interruption, with entire satisfaction and with maximum efficiency?

The maintenance requirements of a filter vary with the type. Filters are designed not only for a particular process of dust elimination, but also for a certain cycle of operation and method of reconditioning the medium. In selecting a filter for a given application, the initial cost should be balanced against convenience and expense of maintenance to insure a satisfactory and economical result.

1. The unit type viscous filter is low in first cost, but its maintenance may be higher than that of the other types, due to the time and labor required for washing and recharging the cells.
2. The dry filter is next in price, with a simplified maintenance. In a widely used dry type, it is only necessary to periodically vibrate and, at longer intervals, change the inexpensive filter sheets.

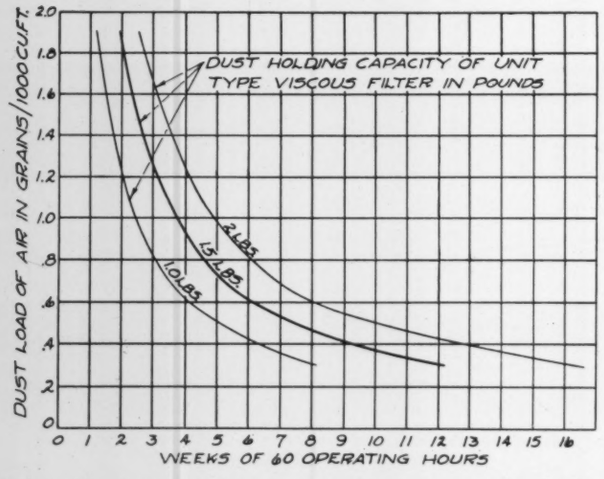


FIG. 1—A TYPICAL MAINTENANCE CHART FOR UNIT TYPE VISCOUS FILTERS

Diagram shows the number of weeks (of 60 operating hours) during which a unit filter will accumulate a dust load of 1, 1.5 or 2 lb. maximum, at various rates of air dustiness.

If the maximum dust load per unit is 1.5 lb. cleaning will be required at the following intervals.

12 to 9 weeks—Rural and Suburban Districts—dust concentration .3 to .4 grains/1000 cu. ft.

9 to 5 weeks—Metropolitan Districts—dust concentration .4 to .8 grains/1000 cu. ft.

5 to 3 weeks—Industrial Districts—dust concentration .8 to 1.3 grains/1000 cu. ft.

3. The viscous automatic filter is high in first cost, but is designed to be low in maintenance. The air cleaning and self-cleaning features are integral parts of the cycle of operations. The filter is automatically reconditioned and maintenance consists of removal of the dirt which is collected in the form of sludge by the filter.

Cleaning Frequency of Air Filters

Since reconditioning of the filter screens, by hand or automatically, is an all-important question in the operation of air filters, it is of interest to inquire into the factors that control the accumulation of the dust. A square foot of filter area in a given installation will require cleaning after having been in service for a certain period of time. The length of this period t is inversely proportional to:

1. The dustiness of air, in grains per 1,000 cubic feet of air. (D)
2. The air volume, in cubic feet per minute. (V)
3. The efficiency of the filter in dust removal (expressed as a decimal) (e)

It stands in direct proportion to:

4. The filter area, in square feet. (a)
5. The dust-holding capacity of the filter medium, in pounds per square foot. (C)

This may be expressed in a simple formula, if a factor is added to establish the correct relation:

$$t \text{ (in hours)} = \frac{Ca\ 116{,}700}{DVe}$$

The effect of the air dustiness on the frequency of cleaning is obvious. For the purpose of our problem, the dust contents of the air to be cleaned may be given by its weight, since at present the storage capacity of filter media is rated in lb. of dust, approximating in composition closely the average atmospheric impurities. Definite values of dust concentration are difficult to establish due to the almost endless variety of conditions which may be encountered. It seems possible, however, to determine working averages for air filters in certain locations, as their operation extends over a long period of time, during which the variations in air dustiness are balanced. This is outside the scope of this article and will be the subject of a later discussion.

The air volume, which is the second factor, determines how much air and consequently how much dust at a given dust concentration will reach the filter surface during every minute of operation. As the filter area is also one of the factors, we will divide the area into the air volume and obtain the "specific rating in c.f.m. per sq. ft." (v) which is equivalent to the velocity of the air when entering the filter medium. The formula would then have the form

$$t = \frac{C\ 116{,}700}{Dve}$$

The ratings of the different filter types have been established by experience. The usual rating of a cell

* Engineer, American Air Filter Company Inc., Louisville, Ky.

type filter may approximate 360 c.f.m. per sq. ft; automatic filters, 450 to 500, and dry type filters, 25 to 50. It is evident that the frequency of service does not depend upon the air volume handled by the filter, or in other words, the size of the filter. The size of the filter determines, however, the amount of surface which is to be reconditioned at every cleaning cycle.

It depends on the efficiency of the filter how much of the dust reaching its surface will be retained by the medium. The effort of all designers of air filters is centered on this point and filters today will trap from 96 to 98 per cent of the harmful dust passing into the filter. As the possible error in the figures for dust concentration is comparatively large, we may set the efficiency of the filter as 1.0, without greatly affecting the result.

The last factor, the dust holding capacity of the filter medium, is intimately connected with the resistance to air flow.

Diagram Shows Cleaning Frequency

The maximum recommended dust load for cell filters of one type varies from 1 to 2 lb. per cell. Using the values of 1, 1.5 and 2 lb., and a rating of 360 c.f.m. per sq. ft. or a capacity of 800 c.f.m. per cell, we can now draw a diagram showing the frequency with which a cell filter must be cleaned (Fig. 1). An operation of 60 hours per week is assumed and the cleaning frequency is shown in its relation to the dust content of the air. It will be seen in a later article that the average dust content of the air in large towns is very close to 0.6 grains per 1,000 cu. ft. For this dust condition, and a maximum dust load of 1.5 lb. per cell, the chart indicates a cleaning frequency of six weeks. An installation, operating under these conditions, may be kept in service and maintained at a constant average pressure drop by cleaning, each week, a sufficient number of cells so that no cell remains in the air flow longer than six weeks.

The cleaning cycle for dry type filters, is as a whole, similar to that of the viscous cell filter. Dry filters must be reconditioned at intervals and the length of service depends a great deal upon the design and texture of the medium. Some of the dry type filters store the dust throughout the depth of the medium in a manner similar to the viscous unit filter. Other types, especially those with renewable media, collect the dust principally on the surface. Their operating characteristics will therefore differ from that of the viscous type. It is best to install a draft gage with every installation and service the filter whenever the resistance reaches a certain maximum, usually 0.5 in. water gage.

Automatic viscous filters operate on cleaning cycles which vary from one week to 24 hours or less, depending on the design. The self-cleaning cycle should be adjusted so that the filter offers a constant resistance to the air flow for average dust conditions. In some cases it is not

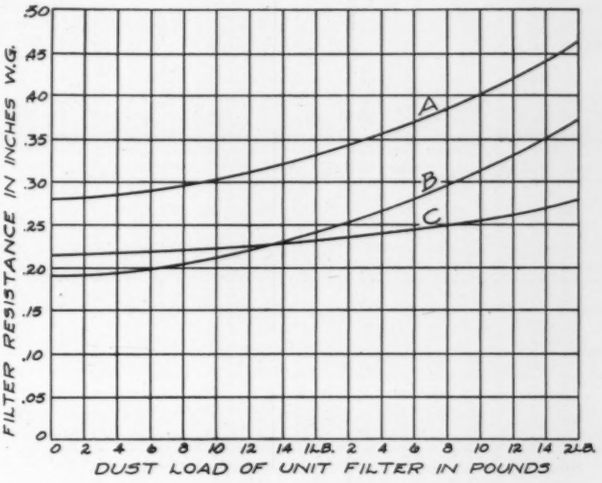


Fig. 2—Resistance Change of Several Unit Type Viscous Filters Due to Dust Accumulation from Tests Made by Author

evident from the start how much dust the filter will have to handle. If the dust condition proves to be severe, the cycle may be changed so that the cleaning function will keep in step with the amount of dust deposited. It is usually preferable to do this with the approval and under the supervision of the air filter manufacturer.

The Resistance of Air Filters

The resistance of an air filter does not increase uniformly as the dust builds up in the filter medium. In most types of viscous media the resistance is almost stationary in the beginning, due to the dust deposited on the baffles in streamline form. Eventually, however, the resistance curve will take on a definitely rising tendency and will finally increase so rapidly that further operation is impractical. Before this point is reached the medium must be cleaned.

In Fig. 2 the resistance of several cell filter types is shown in relation to the progressing dustiness of the cell. The initial resistance of the cell when clean may vary between 0.15 in. water gage and 0.3 in. W. G., depending on the design of the filter medium. A higher initial resistance is offered by cells constructed for the highest possible efficiency, while cells of lower resistance are intended to be used in systems where the available static pressure is limited and a large air volume per cell is desired.

Automatic filters do not usually require the extensive dust storage capacity of cell filters and show therefore a more rapid resistance increase if the dust is allowed to accumulate in the filter screens. For this reason, the maintenance engineer is advised to inspect, at regular intervals, the screens of his automatic filter and make sure that its self-cleaning operation is keeping up with the de-

"..... filters represent an important part of air conditioning systems ..... they will pay excellent return on the investment by keeping grimy, unhealthy dirt out of buildings, plants or processes..... the required amount of care is small and the task simple. It is good policy, therefore, to give the filter the attention it deserves, regularly."

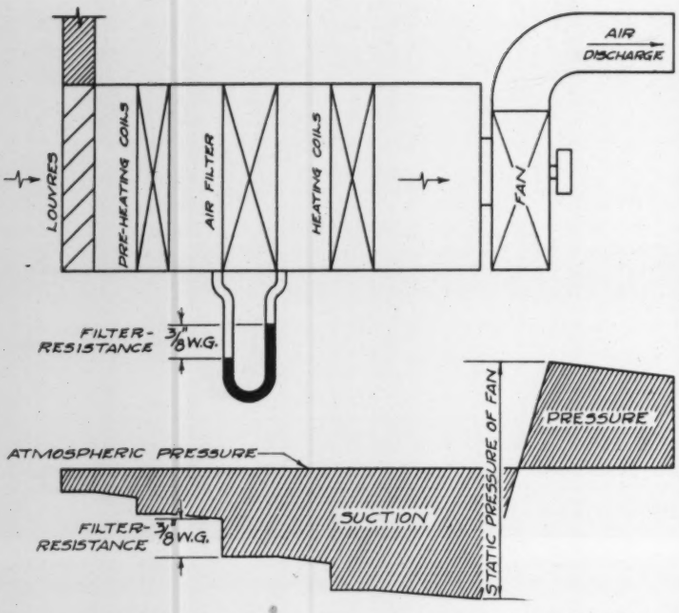


FIG. 3—SCHEMATIC DIAGRAM OF STATIC PRESSURE IN AN AIR SUPPLY SYSTEM

posit of the dust and that the screens are free from partially blocked areas.

It is customary to figure on an operating resistance of 3/8 in. W. G. for a filter installation.

Use of Gage in Operation of Filters

A resistance gage, installed with the air filter, is of great help to the operating engineer in maintaining the air filter in proper condition; an inclined draft gage is often used for this purpose. The tubes leading from front and back of the filter to the gage should be inserted in the casing as close to the filter as convenient. The ends of the tubes should be practically flush with the casing and should stand at right angles to the air flow. So-called "screen nozzles" at the ends of the tubes will give accurate static readings. The diagram in Fig. 3 illustrates in a schematic way the static pressure in a simple fan-filter heating system. It intends to show mainly how a portion of the suction effect of the fan is utilized by the filter and how the gage indicates the exact amount thus consumed.

The static presure necessary to pass the air through the filter must be constantly supplied by the fan and is thus a maintenance expense. The amount of horsepower required by the filter is determined by the following formula:

$$\text{Hp.} = \frac{Vp\ 5.196}{k\ 33000} = \frac{Vp}{k\ 6356} \quad or$$

$$\text{Kw.} = \frac{Vp.}{k\ 8516}$$

V = Air volume in c.f.m.
p = Filter resistance in inches W. G.
k = Fan efficiency (expressed as a decimal)

Roughly estimating the efficiency of a ventilation fan as 65 per cent and assuming a filter resistance of 0.375 in. W. G., the formulas would be further simplified to:

$$\text{Hp.} = \frac{V}{11000} \text{ and } \text{Kw.} = \frac{V}{14750}$$

Varying Operating Conditions

In every-day operation, air filters are subject to considerable changes in temperature, humidity and air dustiness. Variations of temperature and humidity, within very wide limits, have practically no effect on the air filter. Viscous filters for instance, employ a specially prepared charging liquid which retains its fluidity despite temperature changes. To obtain satisfactory operation from this type of filter, only such a liquid developed for the particular filter should be used.

The effect of humidity on filters using dry sheets can be eliminated by treating the filter sheets. Variations of air dustiness are important only if they occur suddenly and are very severe. In industrial plants, for instance, an unusually strong wind may carry a heavy cloud of dust from adjacent sources into the filter inlet. In such a case it is necessary to clean cell filters more frequently than the regular schedule calls for; if automatic filters are used the self-cleaning mechanism may have to be speeded up or run continuously, until the excessive dust load has been removed from the filter medium and normal conditions are restored.

Such occurrences are not very frequent and filters usually have sufficient overload capacity to take care of all usual dust changes.

Maintenance During Construction Period

The maintenance of an air filter should start immediately after completion of the installation. Construction work is usually in full swing when the filters are placed, and mortar, plaster and paint are everywhere and will get into the filter screens unless they are protected. It is therefore advisable to cover all filter surfaces during this time.

Often the fans are operated toward the end of the construction period to dry out the building. Under these conditions the filters will have to handle an amount of dust which is usually much larger than it ever receives after the building is completed and occupied. It is absolutely necessary to prepare the filter for this service and put it in operation before the fans are started. The filter medium of viscous, cell and automatic filters should receive a thorough coating of the charging liquid furnished by the manufacturer. Dry filters should have their filter sheets in place and they should be reconditioned or changed as often as necessary. This will keep the filter in good condition, insure efficient operation and keep the dust from the construction work out of the supply air ducts.

Cleaning of Viscous Cells

The manual cleaning of viscous cells is simplified due to the convenient size of the individual cell. Overall dimensions of a typical cell are 20 x 20 x 4 in. on filters of this type and the weight is usually about 20 lb. This has proved to be an economical size from the standpoint of maintenance, initial cost and proper proportion between effective filtering surface and total frontal area. The cells are held in their frames by simple locks, may be readily removed, and carried by handles which are provided on each cell.

A cell to be cleaned is taken out of its frame and replaced with a spare cell. The dirty cell is then washed

in a tank of hot water, to which a cleaning compound has been added. Special steam cleaning tanks are available which clean the cells thoroughly and uniformly with a minimum amount of labor. It is filled with hot water in which the dirty cell is submerged. Steam is introduced beneath the cell through stationary pipes, setting the water in a rapid motion and keeping it near the boiling point, which washes out the accumulated dirt.

The clean cells are then given a new viscous film by submerging them into a bath of viscous fluid, which is supplied by the filter manufacturer. The surplus fluid is drained from the cell in a warm room, which usually requires 12 hours or longer. The cell then becomes a new "spare cell," to replace other cells in the installation that have accumulated their maximum amount of dirt.

Servicing Dry Filters

Dry filters differ widely in shape, design and structure so that it is impossible to give, in a general way, detailed instructions as to their maintenance. Among the media used in the different makes of filters are cellulose sheets, wool felt, cotton wadding and burlap supported on wire screens. Vibrators are supplied with some of the filters which serve to periodically shake the bulk of the dust off the filter media. In another dry type filter the fabric is kept in motion and is beaten in a dead air space provided in order to clean it. Still other types may be reconditioned by removing the filter cell and rapping it in a specially designed cleaning box. A vacuum hose with special nozzle is frequently supplied with some dry filters, and may be used to relieve effectively the filter from the accumulated dust load. Filters with renewable medium are designed so that the exchange can be made quickly and easily. A sufficient amount of new filter material, either in sheets or rolls according to the design, should always be kept on hand, so that the filter can be refilled whenever it becomes necessary to do so. The soiled filter sheets are collected and burned in the incinerator or under the boiler.

Automatic Filters Should Receive Attention

While no regular daily or weekly chore confronts the maintenance engineer if his filter is of the automatic type, it will be in his own interest, however, not to neglect the filter entirely, just because it is automatic. It is well, as we have seen before, to pay some attention to the condition of the filter screens. The bearings of the fractional horse power motor should be oiled every four to six months. Reduction gears enclosed in housings should be kept lubricated with heavy oil. Chain drives which are exposed to the outside should be greased and all grease cups on filter bearings should be turned occasionally and refilled once or twice a year.

Removal of Sludge

The dirt washed off the filter screens settles as sludge into reservoirs which are usually built into the filter frame. The removal of the sludge is readily accomplished by means of a hand scraper or by removing a sediment pan in the bottom of the filter reservoir. The periodical removal of the sludge is important in filters of the moving screen type. If allowed to accumulate to the point where it touches the screens, the sludge will be carried up in the filter curtain and cause excessive resistance. An amount of oil equivalent to the volume of the removed sludge should be added at each cleaning, to maintain the prescribed level in the oil reservoir. This should be watched very closely, especially in filters with revolving screens. A lowering of the level of more than one or two inches in these filters may allow some uncleaned air to by-pass the filter screen.

FIG. 4—AN INSTALLATION OF DRY TYPE FILTERS SHOWING THE MANUAL VIBRATOR

The sludge from the filter contains, by weight, from 20 to 40 per cent of viscous liquid. To avoid unnecessary waste, one or more large containers of about 20-gal. capacity may be provided, into which the sludge is emptied. Within ten days, and depending on the nature of the dirt, a further separation of the oil and dirt takes place. The clean oil may be poured off the top and used over again in the filter.

Proper Maintenance Pays

Air filters operate under conditions which vary from day to day, and from season to season. They are usually installed in out of the way places, and access to them is often difficult. They do not easily get out of order, as a large margin of safety is built into all their operations. Thus is happens often enough that they are neglected. Consider, on the other hand, that filters represent an important part of air conditioning systems, that they will pay excellent return on the investment by keeping grimy, unhealthy dirt out, that the required amount of care is small and the task simple. It is good policy, therefore, to give the filter such attention as its type calls for regularly.

High Pressure Hydraulic Systems

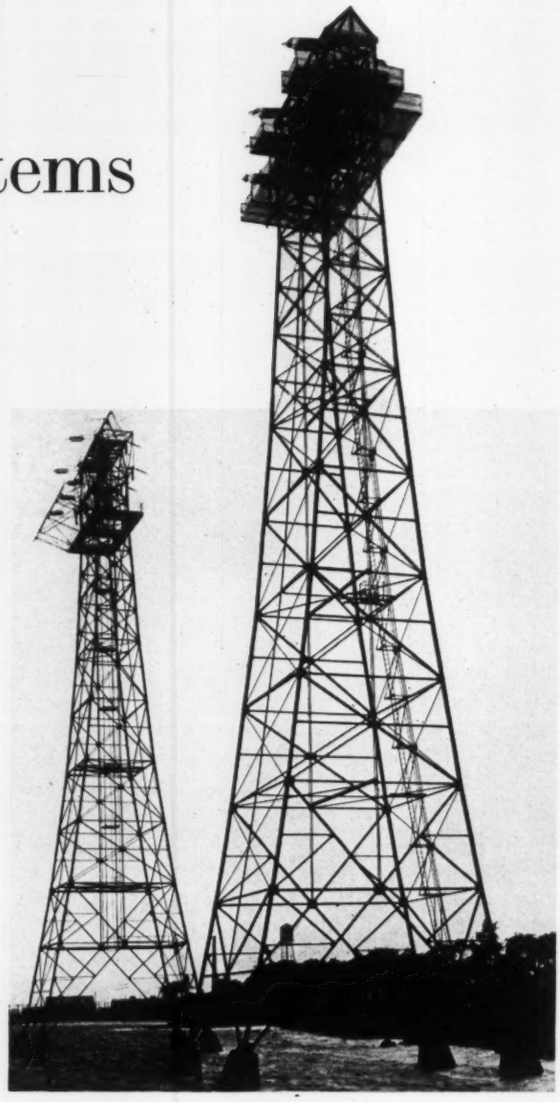
By John H. Ruckman*

THERE is no more marked tendency in modern engineering practice than the utilization of physical conditions of a more and more extreme character. Each year we see higher temperatures, higher voltages, higher vacuums and greater forces. Each year research indicates some new advantage or some new economy which can be obtained by higher pressures. Hydroelectric plants with penstock heads of 1,000 ft. are no longer rare. Oil wells with 1,000 lb. per sq. in. rock pressure are frequent; this figure is common for pipe lines, circulators or cementing pumps. The use of hydraulic presses for forging, forming and extruding metals and miscellaneous plastic substances is steadily increasing as are the pressures used.

It is with the average installation of the last named type that this discussion is concerned, although much of what is said is applicable to the piping which operates hydraulic governors and oil piping installed to maintain pressure in large high speed bearings. In each there is a source of pressure and energy (generally a pump), a system of piping, and a machine of some kind at the opposite end, at which pressure must be maintained and where hydraulic energy is converted into some type of work.

Variation of Power with Pressure

Details of various systems in use differ widely. The power which can be transmitted through a pipe line varies as the pressure, the velocity and the area or square of the diameter. Since the velocity itself varies inversely as the area, the power transmitted for a given quantity of water per second varies directly with the pressure, a condition closely analagous to the increase in electrical power transmitted by a given current, made possible by raising the voltage. Unfortunately the analogy stops here, for whereas the thickness and weight of the electric conductor varies as the current, the weight of the pipe of a given size varies as the pressure. If the pressure and velocity remain constant the friction loss of head in transmission varies inversely as the diameter, while for a given diameter the loss increases with the square of the velocity. Hence for efficiency large pipe and high pressures should be used but the weight of pipe for a given pressure varies as the square of the diameter. It thus develops that the most economical installation will vary for each layout.



TOWERS FOR A HIGH TENSION ELECTRIC POWER LINE

Hydraulic Central Stations

In England high pressure lines sometimes transmit power a distance of twenty miles, with efficiencies above 80 per cent. London, Glasgow and Manchester are all served by central stations, the first named using 750 lb. pressure and the others 1,150. Under conditions existing in that country the controlling factor appears to the use of a design based on the maximum safe pressure for flanged hydraulic cast iron pipe of 6 in. diameter. High head centrifugal pumps are used to supply pressure, and accumulators with heavily weighted pistons are provided to cushion sudden changes in load. The London system in 1911 furnished power to 6,910 machines and used 26,000,000 gallons of water per week (194,000 hp.-hr.).

It will be noted that this type of installation involves low direct operating costs and a moderate charge for physical depreciation. It requires, however, a heavy investment and some danger of loss through obsolescence. This condition is aggravated if peak loads are produced by the occasional operation of one or two very large

* Consulting engineer, Topeka, Kansas.

machines. Capital in Europe generally receives a lower return than in the United States and obsolescence generally is not regarded as so important a factor as in this country. American layouts are generally designed for lower first costs.

Large Plant Broken Up into Three Units

An installation more or less typical of American practice was that of a nitro-cellulose powder plant constructed in New Jersey during the war. The plant covered several square miles but was broken up into three units, an arrangement which saved many miles of piping and transmission losses both for high pressure water, steam and brine circulation lines. The capacity of the hydraulic pressure systems varied somewhat but probably averaged about 50,000 horsepower-hours per week for each unit.

In this case low initial cost was secured by relatively small lines, while efficiency was obtained by shortening the lines (also lessening the first cost) and by increasing pressures. The energy in this case was provided by multiple piston gear-driven pumps. Initial pressure varied but was generally carried at 3,500 lb. per sq. in. Pressures at the machines varied somewhat, particularly at points distant from the power house, but generally it was held above 3,200 lb. There were some seventy presses of various types operating from a single central station and when these temporarily fell "in step" there was a noticeable but not generally serious loss of pressure. Piping was of double extra heavy lap welded steel, the largest mains being 4-in. Ells were avoided by long bends in the pipe, while the layout was such as to minimize the necessity for tees. Connections were made by screwed flanges. All valves in the high pressure system were either angle or globe valves with conical seats. All fittings were of forged steel.

With very large single machines handled by independent pumping units, the largest quantities of water are pumped the shortest distance, generally. Accumulators may be dispensed with, or in the case of machines of intermittent action, an accumulator may be provided and the size of the pumps reduced.

Control Valves

In all systems the control valves of the various machines represent an important problem. In a small machine, with rapid action, valves can be thrown wide open or tight shut rather suddenly. The small size permits the use of valves of the plug cock type which can be ground to fit snugly and can be kept tight without involving undue friction. As a result, wear is small. Where rapid action is desired in moderate sized machines balanced and piston valves are sometimes used, with packed plungers (often cupped leather).

Where slow, modulated motion is desired, some type of long stemmed globe or angle valve with a conical seat may be employed. Where very large units are to be controlled it is generally necessary to operate the valves by auxiliary hydraulic or electrical devices. Several types of valves are used, but it is the writer's belief that there is still room for improvement. The conical nozzle control valve generally used today in the penstocks of high head impulse wheels has some very excellent hydraulic characteristics which should be applicable.

Sudden closure, of course, produces water hammer and in large machines this is an additional reason for decreasing the length of the pipe system to a minimum. At a pressure of 2,000 lb. per sq. in. the velocities in the throat approach 547 ft. per sec., and even where the fluid is entirely free from grit serious wear may occur. As a result, except where carefully purified oil is used as a fluid, it is difficult to keep control and modulating valves tight. The use of alloys renders points and seats resistant to chemical action of oxygenated water but some grit or scale is nearly always present in the system and this material may groove both points and seat. For this reason a control valve can never be depended upon as a cut-off for any machine. Another valve must be provided on the pressure side, which should never be left in any position except wide open or tight shut, and except in emergency it should be moved only when the control valve is fully shut. It should be opened and closed as rapidly as possible.

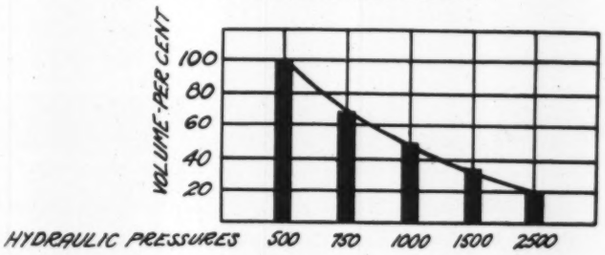
The Question of Leakage

The matter of leakage is one of the chief problems which face the maintenance engineer, and the operating personnel. So long as valves, press pistons and other moving parts function as designed no trouble occurs, but where there is motion there is friction and where there is friction there is wear. Once wear has taken place leakage starts and when once started it develops at a rapid pace. This is particularly true when piston leathers give way. The writer has seen a press apparently functioning perfectly develop a leak and in three minutes become useless. Where the machine involved is an extruding press this may be a very serious matter, as the material behind the dies may become hard through cooling, "curing" or solvent evaporation, and may have to be removed at great expense.

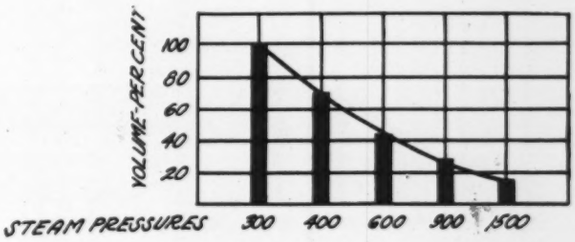
Causes of Shut-downs

Hydraulic machinery is relatively expensive, and as a result a plant is seldom equipped with any more of this type of apparatus than is absolutely necessary. As a result, the hydraulic units tend to become operating "bottle necks," the capacity of the plant is likely to be

Relative Volume of Fluid Required for Equivalent Available Energy in Hydraulic Systems



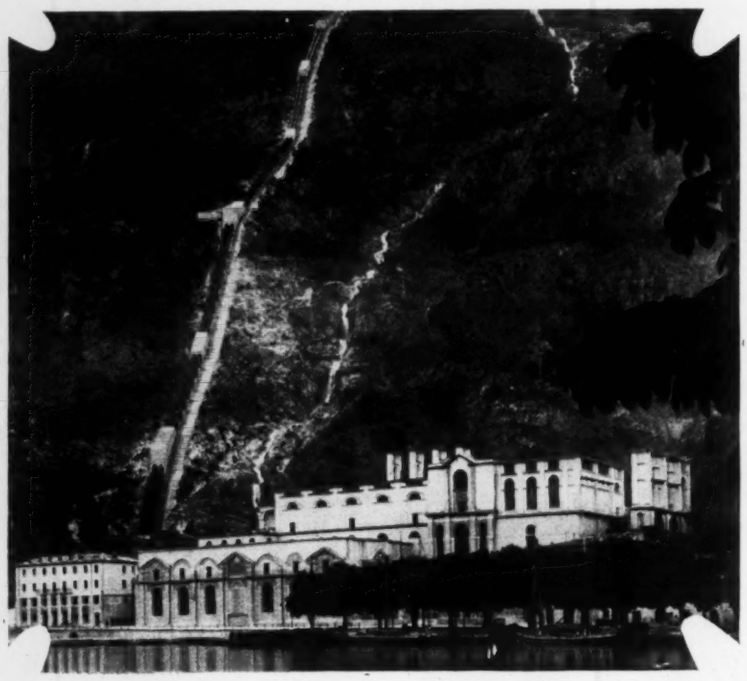
Relative Volume of Steam Required for Equivalent Available Energy in Modern Steam Power Plants



that of its hydraulic equipment and operating delays are expensive in the extreme. Since the equipment is, generally speaking, of simple and rugged construction, outages are usually caused by three factors: failure in the primary source of power, accidents, or leakage. The first difficulty is outside the scope of this discussion but the accident and leakage demand attention.

Accidents

Accidents, of course, should never occur, but they do even in the best managed plants. They usually happen through faulty design, errors in operation or deficiencies in maintenance. The power of high pressure water to do damage is not generally appreciated by those who have not witnessed it at work. Many suppose that because water is nearly incompressible it has no power to "follow through" in case of a break. This is far from being the case. If an accumulator is present it will discharge its entire energy. Water at atmospheric pressure will absorb approximately two per cent of its volume of air. Under pressure it absorbs additional volume in proportion to the additional pressure. In a hydraulic system the amount of air in solution depends solely on the opportunity of the water under pressure to come in contact with air. In any ordinary system the amount is always sufficient to expel water for several seconds. Even if an entirely gas-free medium is assumed, sudden contraction of the steel piping or press barrels will expel about one-tenth of one per cent of the entire water in the system. In a line with an internal diameter of 3.152 in. and a mile long this amounts to some two gallons only, but even this will weigh over 16 pounds and will develop a speed of 600 ft. per sec.; amply enough energy to drive a bolt or valve stem through a one-eighth inch steel plate, and quite capable, without any metallic missile, of literally tearing a man to pieces should the jet strike him.



An Italian Hydro-Electric Plant. Note the Penstocks

For this reason considerable care should be given to so arrange valve stems and flange bolts so that in case of failure they will not be thrown directly toward points commonly occupied by operatives. Commercial high pressure valves today are so designed that the stem can not be blown out through the stuffing box. Nevertheless, the writer has frequently noted "home made" operating valves which were not protected in this way, and he has on several occasions seen such valves opened too wide. The result was generally a clean round hole in the roof, sometimes a broken wrist for the operatives, and always a flooded building with from fifteen minutes to an hour's delay and serious loss to materials in the process. In extrusion presses, dies and even die plates may become similar missiles.

Another source of accident lies in the operating release valves. When the ram is drawn back they are opened and should tend to stay open. The throb of the pumps, and the jolt of other operating valves may permit the lever's weight to act, and should this tend to close the valve it may cause serious consequences.

On one occasion, a single operative was running an extrusion press. Through some error in a previous process the charge hardened in the barrel and he went up to the charging floor to remove it by hand. He left the ram up, high pressure discharge valve open and the high pressure operating valve closed. He did not close the cut-off valve, and the operating valve leaked. While he was working the discharge valve shut, permitting pressure to build up on the ram. The first warning the unfortunate pressman had was the discovery that he could not remove this head from the barrel and assistance arrived just too late.

Yet another dangerous situation occurs where faulty packing or an error in manipulation of valves permits high pressure water to reach the low pressure system. It would seem common sense to equip all low pressure systems with pressure relief valves, but this precaution is sometimes omitted. The writer recollects two incidents in connection with an oil field installation. In one case, as the result of a misunderstood signal, the low pressure piping burst; in the other, as a result of a slow leak in a cross connection, a low pressure water meter exploded, killing a meter reader who had no warning whatever of his danger.

Piping Outside the Plant

Accidents to the high pressure lines outside of buildings where the mains have been properly laid are rare. Freezing may cause trouble, particularly if the plunger of the accumulator is frozen or stuck from any other cause. Properly calculated shearing pins or electrical over-load switches properly set will minimize damage in gear-driven layouts and pumps can be designed so that they will refuse to deliver against a dangerous head.

Damage to Piping Inside the Building

Inside the building, however, piping may be damaged. Cranes carrying heavy loads may be dangerous; industrial power truck and derailed narrow gage loco-

motives sometimes hit a high pressure main. For this reason, if for no other, piping should be kept, so far as is possible, below the floor. Where large forging and flanging presses are used, internal piping which will not be injured by accidental shocks from billets or plates is a considerable advantage. Needless to say, a burst main or a blown gasket in a foundry is a disaster. Whether the water strikes partly completed molds or hot metal the loss will be serious, and in the latter case, of course, injury to personnel will also occur.

Gaskets

Gasket failures occasionally result from deterioration because of the heat of adjoining steam pipes or because of leakage of oil. High pressure piping in or near cupolas or melting furnaces should usually be set up with metal gaskets.

Wear on Cylinder Wall

It has been noted that some accidents and many delays result from wear and leakage. Wear is particularly important in stuffing box and valve bonnet screw threads, in pins and bushings of interlocking safety devices and in rams, pistons and cylinders. Every time a piston leather fails, the wall of the cylinder is slightly eroded and the rough spot thus created tends to wear the next leather in the same place. Slowly the entire barrel becomes grooved and roughened and leathers can no longer be made to hold long enough to permit the machine to operate. The cylinder must then be rebored and a new piston fitted. The cost of dismounting, moving, setting up, machining and remounting one of these cylinders with its piston is very serious; a logical method would seem to be the development of a relatively light boring machine which could be aligned to the piston barrel and which could refinish the barrel in place. This general method of machining heavy castings has received wide application in the manufacture of hydraulic machinery but has not as yet been adapted to maintenance work to the extent which is possible. Another system which offers possibilities is that of removable liners. Its practicability has been demonstrated in connection with moderate-sized cannon but the machining of liner and outer bore must be precise.

Definite System of Maintenance Needed

Enough has been said to indicate the necessary steps for proper maintenance. Wear is constantly going on and leakage is constantly developing. Obviously, worn parts must be replaced and leaks stopped before they become severe. As in every other line of maintenance work, this can be accomplished only by a definite system of testing, inspection and replacement based on accurate records.

Opportunities for Inspection

Even where plants operate on a three-shift basis there are generally one or two shifts each week during which most of the machinery is shut down. If there is absolutely no shutdown it is still possible to arrange with the operating force for certain tests. One of these is that of closing all cut-off valves at a given time, shutting down the pumps and noting whether the accumulator piston falls, or as a more delicate test a microphone can be applied to the outgoing main.

In very large systems this test is nearly impossible and as a result there may be undetected leakage. In the moderate sized plant, such as is typical of American practice, there should be little leakage. If there are certain units that can not be shut down these units should

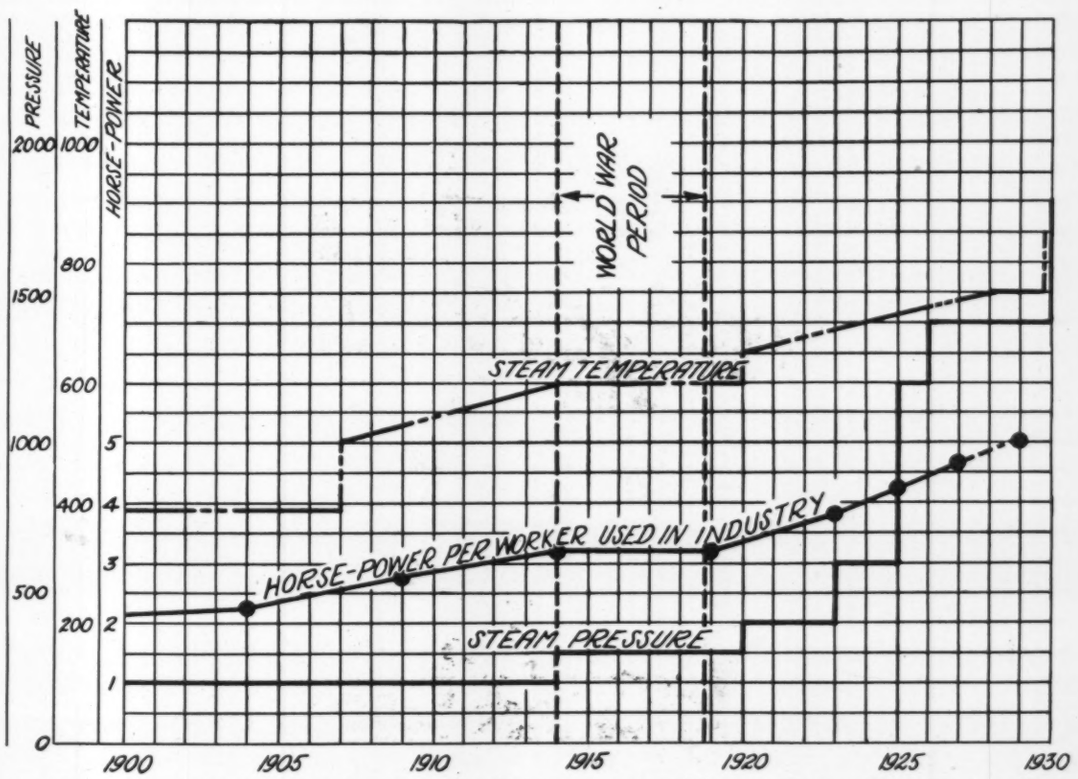


Chart Showing Rise of Horse-Power Used in Industry and of Working Steam Pressure and Temperature

be tested to determine their exact consumption of water so as to compare this consumption with the output from the accumulator. Where a considerable shut-down period is allowed there should be a regular system of testing the operating valves. If shut-downs are not sufficient to permit this, continuous inspection must be made. By the use of a microphone or stethoscope the inspector can listen to the action of the various mechanisms and can follow the development of leaks from their inception.

The mains should be regularly patrolled. If placed in conduits the drainage from each section of the conduit should be concentrated at a certain point to permit the amount to be observed and to give the earliest possible notice of an incipient failure.

Records of Value

These methods give the maintenance force a very good check on what is going on but they do not prevent shut-downs at awkward moments, for when a leak once starts it develops with startling rapidity. It is at this point that records become of value. It is a well recognized principle in taxicab, motor truck, airplane and railway locomotive maintenance that a given part or a given machine can be depended on for a certain number of hours performance and that it should then be overhauled no matter how perfectly it appears to be functioning.

This same system can, in certain cases at least, be applied to hydraulic valves and mechanism. The writer realizes that this is a subject on which there is an honest difference of opinion. For instance, most maintenance foremen will declare flatly that the length of life of a set of piston leathers can not be predicted and that the less they are "monkeyed with" the better.

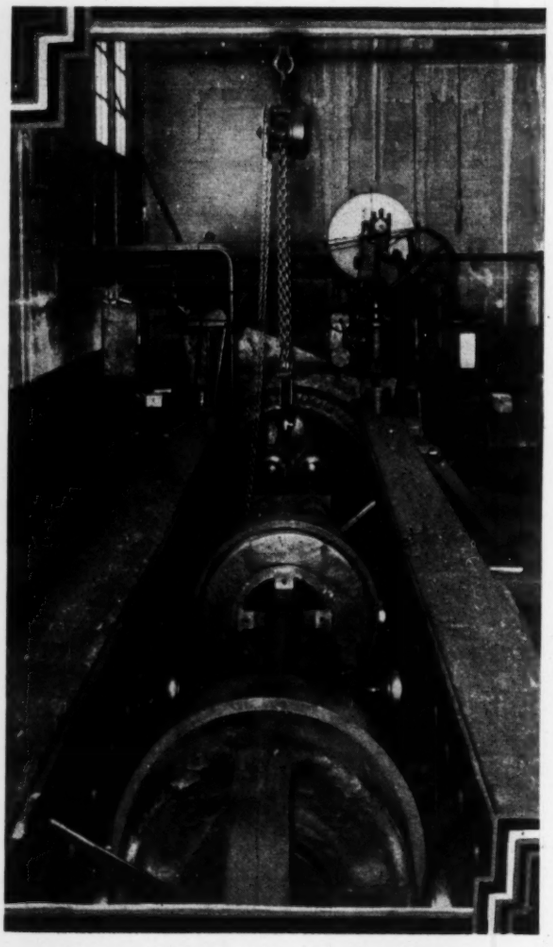
There is enough truth in this statement to warrant consideration but it is the writer's observation that the first set of leathers installed in a true and smoothly machined cylinder last a long time. The second set, if placed before the first set has worn out, may last even longer, but if for any cause a leather is permitted to remain in use after a strong leak has begun to develop, the life of subsequent leathers in that particular barrel is liable to be very uncertain.

If accurate records have been kept relative to replacements, it becomes possible to analyze these and determine to how great an extent a system of anticipating replacements is likely to succeed.

Where the machines are heterogeneous in type (as frequently is the case in boiler works, shipyards and specialty factories) it is difficult to devise any very set system of replacement. Where there are a large number of similar units, repair work can be reduced almost to the precision of a drill, the maintenance men being supplied with specific equipment and following a predetermined procedure. Where this is possible very efficient work can be performed rapidly. Saving in time of repairs can also be accomplished by working out a system of unit replacements. The nature of such a system will differ for each individual plant but in instances where the machines are allowed only brief shut-downs, it is often easier to replace an entire assembly than to discover just what part is giving trouble. The details of such a scheme must be worked out with due consideration to cost of delays to machine, labor cost and time of repairs, and cost of carrying additional repair parts.

Fluid Must Be Pure

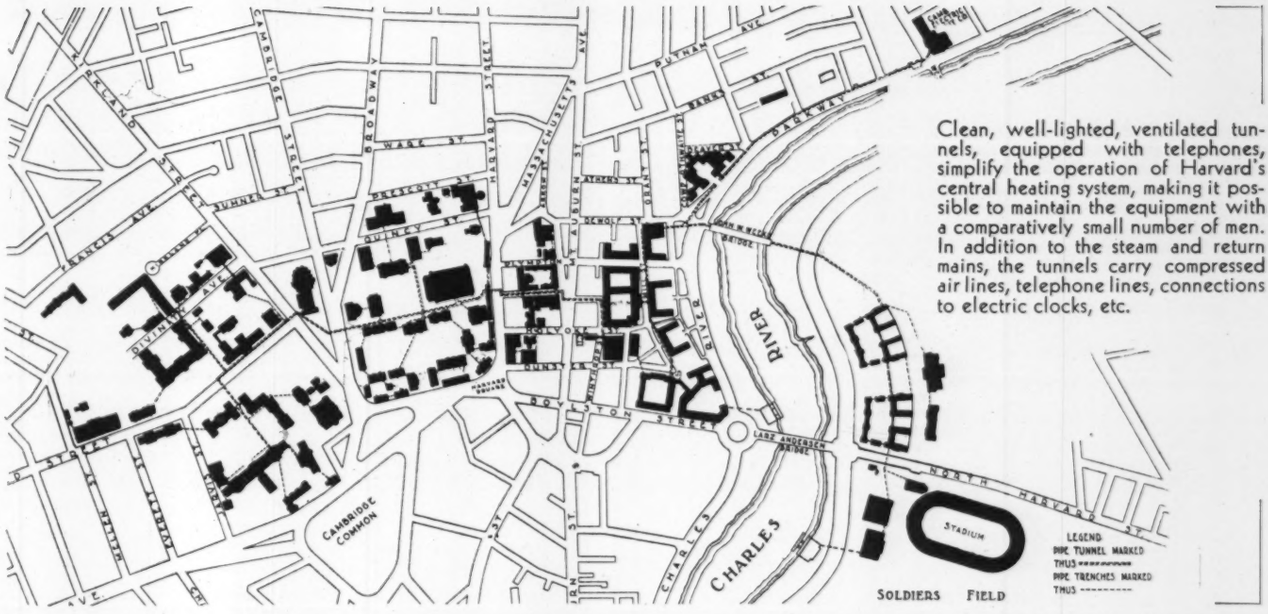
Another vital preventive measure is the careful supervision of the purity of fluid supply. Oil in a closed system becomes dirty very slowly and is seldom gritty. Water drawn from wells or rivers is invariably contaminated with grit, some of which is of microscopic size. This must be carefully settled and filtered out. Even when the water has been cleared of grit the problem is not solved for water (unless pre-heated) will carry oxygen in solution and this may react in pipes to form iron oxide scale. Some of this scale is likely to take the form of hematite or magnetite, both hard minerals which when carried at high speeds past valve seats, or pistons, develop enormous cutting power.



600-Ton Hydraulic Pipe Testing Machine

The use of the hydraulic principle is popular. We are utilizing it in presses, shears, flangers, bolt headers and riveters, and for feeds, chucks, governors, stokers and remote control. Tendencies in America seem to be away from the central plant, with transmission of power in the hydraulic form. We prefer to transmit power as electricity and use the hydraulic principle as a machine element.

Whatever the nature of the installation the same rules of operation and maintenance hold. Emergencies must be met before they occur and repairs and replacements must be made when and under conditions chosen by the plant executives, and not dictated by the uncontrolled action of corrosion and erosion.


Clean, well-lighted, ventilated tunnels, equipped with telephones, simplify the operation of Harvard's central heating system, making it possible to maintain the equipment with a comparatively small number of men. In addition to the steam and return mains, the tunnels carry compressed air lines, telephone lines, connections to electric clocks, etc.
RIVER
CHARLES
CAMBRIDGE COMMON
STADIUM
SOLDIERS FIELD
LEGEND
PIPE TUNNEL MARKED THUS
PIPE TRENCHES MARKED THUS

A Plot Plan of the Buildings at Harvard University. Gravity or Forced Circulation Hot Water Systems Are Used, Depending on the Size of the Individual Building. Convertors in the Basement of Each Building Are Supplied With Steam From the Plant of the Cambridge Electric Light Co.

How Harvard Heats a Hundred Halls

By G. K. Saurwein*

THE supply of heat to Harvard University, with more than 100 buildings—a total volume of about 70,000,000 cu. ft.—is accomplished through a system of trenches and tunnels, connected to a central steam plant.

Until about fifty years ago, the students of Harvard University, which is soon to celebrate its 300th anniversary, took care of their own heating requirements by burning wood or coal in the fireplaces of their rooms, just as they provided their own water from the pump in the college yard. While all of the rooms in the new housing units are still fitted with good fireplaces, to lend a cheery and homelike atmosphere, the heating requirements are now met with modern radiation, both direct and indirect.

A Variety of Systems

It can readily be imagined that an institution so long in the making has a wide variety of equipment in its buildings. As new buildings were added they were equipped with various types of heating systems, and so there are warm air systems, one pipe and two pipe steam systems, vacuum systems, and hot water systems, both gravity and forced circulation. Over 30 years ago, a forced hot water system was installed in Pierce Hall. The piping in this system has recently been examined and found to be in first class condition, although the pump and convertor have had to be reconditioned.

Hot Water Systems Now Used Extensively

Recent installations have been hot water systems with convertors located in the basements and supplied with steam from the tunnel mains. Gravity circulation is utilized in buildings of say 750,000 cu. ft. volume; in buildings of larger size forced circulation is used, depending somewhat on the arrangement of the structure. Indirect heating is fed with steam, the equipment being located in the basement with the convertors and pumps for the direct heating, where it is easily accessible from the tunnel.

The Old Central Steam Plant

At first each building had its own heating plant. Later various groups were tied together and heated from the central building in the group. In 1914 the construction of the present extensive system of tunnels was begun. Steam was supplied from the nearby reserve station of the Boston Elevated Railway, which the University has lately purchased. These tunnels were extended until now nearly all of the buildings contiguous to the yard, from the Charles River north to Langdell Hall, are connected. In 1927 twenty-two boilers were removed from buildings which were added to the system. When the present construction program is completed, which will be in about two years, the total connected volume will exceed 75,000,000 cu. ft.

The old Harvard steam plant of the Boston Elevated Railway Company, located on the river, containing six 4,000 and four 6,000 sq. ft. boilers operating at 180 lb.

* Engineer, Harvard University.

ABOVE — DETAIL IN A TUNNEL SHOWING ANCHOR AND EXPANSION PIECES

RIGHT—DETAIL SHOWING PIPE FABRICATION IN A MACHINE ROOM

delivered saturated steam through reducing valves at about 100 lb. pressure, which was sent north through the tunnels in two 10-in. mains and across the river under the Weeks Bridge to the Business School and Soldiers Field in a 10-in. main. This steam plant did not have sufficient capacity for the contemplated additions to the system. Fuel was brought to it over the street railway tracks. The boilers were 22 to 32 years old; they were set low, fired with stokers. Little could be done to increase capacity without building almost an entirely new plant. Therefore, since the site was better suited for other purposes, it was decided to abandon this plant.

Steam Now Supplied by Utility

Plans were made for a new plant to be located on Western Avenue on the fringe of the University development, where rail delivery of fuel was possible. This plant, however, was never built because negotiations with the Cambridge Electric Light Co.—begun at about that time—resulted in a very favorable contract for the supply of steam from its nearby plant. The company offered to supply steam at nearly the same cost which would result if the new University plant had been built. This, together with the fact that the Corporation desires to do business with the local utilities whenever conditions are favorable, led to the decision to enter into the contract.

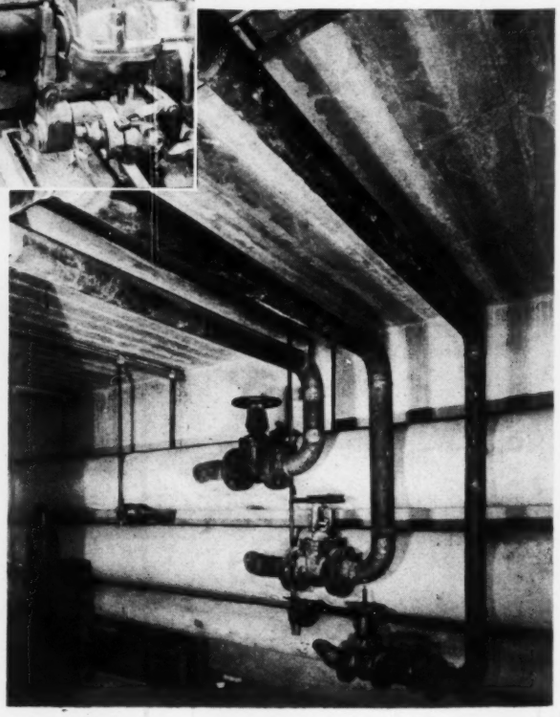
The Tunnel System

It was necessary then to build a connecting tunnel about 2,000 ft. long between the steam plant and the existing tunnel system. A tunnel of approximately the same length would have been required to the proposed new plant, but not in so favorable a location. The tunnel as built runs along the Charles River Parkway, its top being a 10-ft. curved granolithic walk; it carries two 12-in. steam mains and one 8-in. return. It is located in a developing area, already serving a large housing unit with the probability of additional connections.

The main tunnels are 7 ft. high and for the most part 8 ft. wide, branch tunnels being 5 ft. wide. In addition to the steam and return piping, they carry a compressed air line at 50-60 lb. pressure for operating automatic heat control equipment, sewage ejectors, etc.; the pressure regulators and shut-off valves are located there; the tunnel telephones and the telephone cables connecting the various buildings to the University private branch exchange are run through the tunnels, as are also the interconnections to the electric clocks.

Most important, they serve as passages between buildings

BELOW — DETAIL IN A TUNNEL SHOWING A BUILDING CONNECTION



and into the apparatus rooms of the various buildings for the operating personnel who control the heating service and attend to the equipment. This arrangement, with the tunnel phones, makes it possible to operate with a comparatively small number of men, as one man can take care of several buildings.

Steam is supplied at the plant at 100 lb. pressure and 50 F superheat. In moderate weather one main will serve; in severe weather, both are required. Pressure regulating valves and shut-off valves are located in the tunnel at the buildings. The steam pressure is reduced to about 5 lb. for heating. For kitchen use about 25 lb. is supplied. Heat is shut off at the buildings in the evening and turned on in the morning in accordance with a schedule which varies with the use of the building and with the outdoor temperature. In zero weather, the steam is kept on all night.

Returning Condensate Saves 10 Per Cent

Between ninety and ninety-five per cent of the condensate is returned to the boiler plant at a temperature of about 160 F. This effects a saving to the University of more than ten per cent, for the cost of heat is on a Btu basis. Lifting traps are largely used for returning the condensate to the return main, their exhaust being piped back into the collecting tanks. In some of the newer buildings compressed air is used satisfactorily for operating lifting traps. At the larger centers of collection, centrifugal pumps are installed for returning the water.

Recording-integrating flow meters, steam temperature recorders, and a V-notch condensate meter with a temperature recording pen provide the data for the determination of the heat supplied by the plant and the charge therefor.

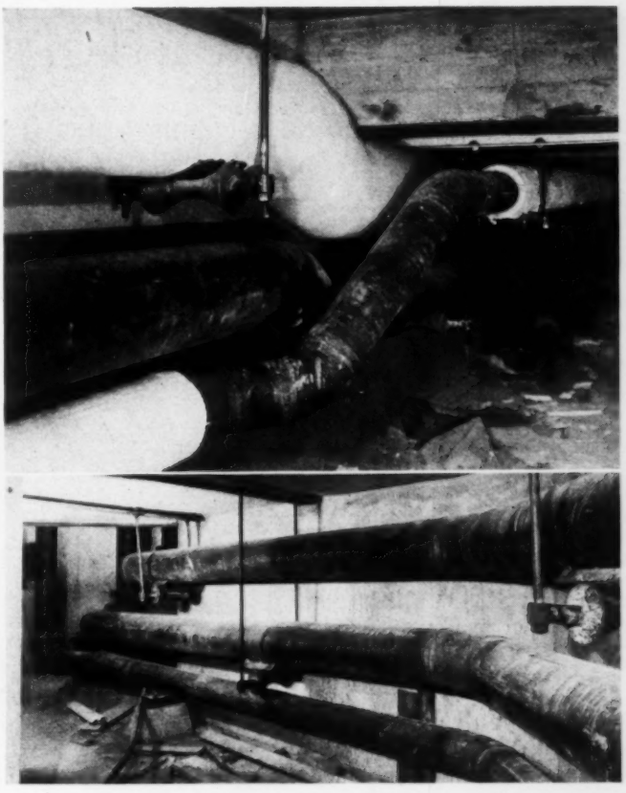
Distributing Heating Charge to University Departments

For the purpose of distributing the charge among the departments of the University, the lifting return trap has been calibrated by means of a by-pass connection and weighing tank, and fitted with a counter to register the number of dumps. The approximate quantity of condensate may then be calculated. The accuracy of this means of measurement depends upon the perfect seating of the check valves in the connections. Leaking check valves at these pressures introduce errors of varying magnitude. In the later installations standard condensate meters have therefore been provided. For the larger users, at locations where the line pressure is fairly constant, the electric steam flow meter is now being used.

Insulation of the Piping

Expansion is taken up in wrought steel slip joints having double anchor brackets, mounted between vertical channels. These joints and the main valves are flanged and bolted into the lines. All other pipe joints are welded. Welding elbows are used entirely. The steam lines are covered with double-thickness magnesia, the slip joints having demountable covering sleeves; the return lines have single-thickness covering. The insulation work is done thoroughly so that the heat loss is low.

The tunnels are clean, well lighted and ventilated by means of fans, the warm air whenever possible being blown into the basements of buildings where heat is required.



Two Photographs Taken During the Construction of the New Central Heating Tunnel. The Tunnels Are Ventilated, the Warm Air Being Then Blown Into the Basements of Buildings Where Heat Is Required

"Heating and Ventilation" Revised

The third edition of Allen and Walker's "Heating and Ventilation" has been thoroughly revised to bring it up to date with the many advances made in the field during the past few years. Information on industrial air conditioning and air cooling, drawn from several sources, has been included. Another feature of the new edition is a chapter on hot water heating systems, which was prepared by Professor Giesecke, an outstanding authority on this type of heating.

While intended primarily as a college textbook, or a volume for individual study, this work has also found considerable favor as a reference for the designing engineer. Emphasis has been placed on the discussions of principles rather than methods, though the latter are not slighted.

Chapter headings include heat; heat losses from buildings; methods of heating; warm air furnace heating; properties of steam; radiators; fuels and boilers; steam heating systems; pipe, fittings, valves and accessories; steam piping; hot water heating systems; temperature control; air and its properties; ventilation principles; systems of ventilation; design of fan systems; unit heaters; air cleaning and humidifying; artificial cooling; industrial air conditioning; and central and district heating —electric heating.

Published by McGraw-Hill Book Company, Inc., 370 Seventh ave., New York City, this text is 6 in. by 9 in. and contains 426 pages. It is cloth-bound; the price is $4.00.

"Open for Discussion"

How Should Heating and Air Conditioning Terms Be Revised?

A Fan Engineer's Thoughts

By W. A. Rowe*

IN considering the revising of heating and air conditioning nomenclature,[1] probably some comments on specific recommendations would be in order. In the first place, *power* is the time-rate of doing work. It may be expressed in watts, joules, or horsepower, depending upon the units involved. It is very necessary in fan engineering to determine the horsepower as it is required in order to purchase suitable drive equipment. We find that for a given horsepower on a particular unit the motors we purchase will have widely varying watts input, depending upon the efficiency—and with alternating current, the power factor—of the motor. Sometimes the watts input will vary 50 per cent between different motors for the same actual horsepower load of the fan. I am afraid we can't abandon the use of horsepower until we select some other arbitrary unit to replace it.

I question the substitution of the word "draft-head" for "flue-height." I see no objection to the latter, as it indicates exactly what the word conveys. You could have the same flue-height for the various types of heaters, and yet have a much higher draft-head when measured on a draft gage for one which gave higher temperatures than with the other.

The suggestion to use the term "thermotrap" is substantially what the manufacturers have been using more and more, namely, "thermostatic trap." The suggestion of Mr. Lewis' has the merit of abbreviation.

With regard to the definitions of different types of steam heating systems, it does not seem to me that the designations suggested are comprehensive enough, or distinguished between them sufficiently to cover all the types of systems in every day use. It does not seem to me necessary to have a pump to have a vacuum system. You could have a system functioning a considerable portion of the time on a partial vacuum produced by means other than a pump. Furthermore, it does not seem to be necessary to have a thermotrap to indicate a vapor system. What designation would Mr. Lewis have for a Mouat vapor job?

The reversion to the old dictionary in the terms insulation, valves, dampers, register, and air filter is most refreshing.

"Bulb chamber" does not seem to me to be as suitable as a "thermometer well."

I agree with Mr. Lewis one hundred per cent on obsoleting the arbitrary "square foot" of radiation, which seems to be coming more out of place every day and is no more logical than, for example, the equivalent lineal feet of pipe coil.

I have deliberately left until the last, discussion of his newly coined words for unit ventilating and heating equipment. If new designations are to be adopted, I think they should be in current phraseology using present well known words. There does not seem to be much question of what is meant when a device is described as an "electric driven gas fired unit heater," or a "unit ventilator," leaving out the "electric" prefix if you will, as so much unit equipment nowadays is using that method of drive.

The use of newly coined words should be left to the

* Chief engineer, American Blower Co.

[1] *Suggested Reforms in Heating and Air Conditioning Nomenclature*, by Samuel R. Lewis, HEATING, PIPING AND AIR CONDITIONING, September, 1930, p. 744.

Last September, an article by Samuel R. Lewis was published, suggesting the need for a standardized and universally used list of terms for heating and air conditioning equipment and systems. In this department are published two comments from other engineers (such comments have been published from time to time) and a resumé by the the author. Below are shown the terms suggested in the original article.

GLOSSARY OF NEW TERMS SUGGESTED

(1) Power—Horsepower. Let us do away with the horse.
(2) Elecon—Recirculating fan-equipped unit heating machine.
(3) Elevent—Unit ventilating machine which takes its supply of air from out-of-doors.
(4) Fluecon — Gravity cabinet convection heater; any heater placed in a flue and recirculating by gravity.
(5) Fancon—Blast indirect heater.
(6) Draft-head—Vertical distance from the top of the fluecon to the bottom of the outlet opening for the air which passes around the fluecon.
(7) Thermotrap — Thermostatic trap used with vacuum systems or with vapor systems.
(8) Direct-Vent System—Any steam heating system which uses air valves which discharge directly from the heater to the atmosphere. Let us call the single-pipe system and its name both obsolete.
(9) Vacuum System — Any steam heating system which uses thermotraps and a pump.
(10) Vapor System—Any steam heating system which uses thermotraps and gravity return of the condensation to the boiler or to the receiving tank.
(11) Insulation—To be used instead of "covering."
(12) Valves—Used on pipes but not in ducts.
(13) Dampers—Used on ducts.
(14) Register—Combination of register face and register body.
(15) Register Body — Dampers and body which usually are attached to the face.
(16) Bulb Chamber—Thermometer receptacle for ducts or pipes.
(17) Air Filter—Air cleaning device, as differentiated from a dust filter.
(18) Square Foot—An obsolete term.

individual manufacturer on which he may obtain copyright protection. This the manufacturer can not get on words which are already currently used and well known, and the value of a coined word such as "kodak" is well known. In this respect Mr. Lewis' services would probably be more valuable to the individual specialty manufacturer than to the profession.

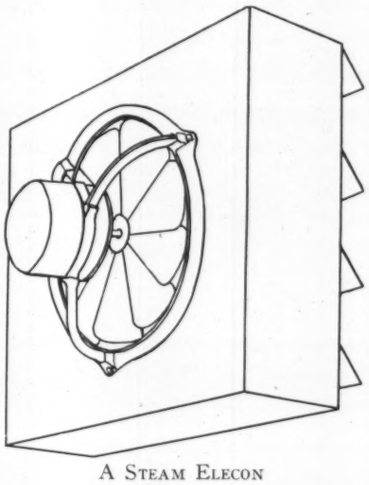
The Jobber's Salesman's View

By D. Fraser Michie*

SAMUEL R. LEWIS, in his article on reforms in heating and air conditioning nomenclature, invited discussion on the subject. Accepting this invitation, I want to make a few comments from the viewpoint of the jobber's salesman.

Some of the suggested new terms provoke no comment from me. On those where our ideas differ, my comment is as follows:

Unit Heater—A term fairly well established by custom; generally understood by both trade and public. The derivation of the coined word "elecon" is clear *when explained,* but not as self-evident as "unit heater." "Elecon" is a word in a foreign language to the uninitiated, a bad feature from the standpoint of the man talking to the average user, who should be considered. I am for keeping the "unit heater."



A Steam Elecon

Unit Ventilator—So here, as with the "unit heater," keep—as *E. B.* Lewis[2] says—the "unit ventilator" and the "unit conditioner."

Cabinet Convectors—Certainly "concealed radiator" or "encased heater" are not satisfactory designations. However, I don't like "fluecon" because, again, we have a coined word which must be memorized in order to know what it means. I suggest "direct convector," recessed or exposed, which seems proper enough and clearly descriptive.

Blast Indirect Convector—The present terms in use are very poor and should be changed, but why "fancon" if a fan is not always used? The derivation is inaccurate here as compared to "elevent," "fluecon," etc. My suggestion is "indirect convector," with the fan or induced draft part of the specification given separately.

Draft Head—I'm agreed that this is an acceptable term. E. B. Lewis (in his discussion) seems to think "heat-head" better because, customarily, "draft" makes one think of a chimney, but it just as often makes one think of air currents in a room (although usually of cold ones!), and for this reason is properly applicable. Of course, the meaning of any term adopted should be standard; top of heater to bottom of outlet seems to be it.

Direct Vent System—Or "direct vent vacuum system" is fine, broad enough and quite clear.

Vacuum System and Vapor System—These would be clear if made a standard and adopted by engineer and manufacturer. The trouble would be in getting the manufacturer to change the name of his pet.

Register—And "register face—body—dampers." What do they register? Surprise? Rather "grille" (for the complete assembly), "grille face," "grille frame" and "grille dampers."

Steam-supplied Hot Water Heater—"Water tube" and "steam tube" are very clear, but "heater" might better be "exchanger" to get away from "heaters," which are coal, gas or oil fired.

Square Foot—Here we have plenty of room for improvement and for argument about what such improvement shall be. We all apparently agree that "square foot" is obsolete, but not that "Btu" is the proper successor. E. B. Lewis, with the metric system in mind, suggests "therm" (100 Btu), "centitherm" and "kilotherm." To be strictly metric it would be better to make the "therm" 1,000 Btu so we could use the prefixes "mille" and "kilo" more strictly. The 100 Btu "therm" would then be a "centitherm." This gives us a convenient measure for various purposes—the "millitherm" (Btu's by 1's) for expressing constants; the "centitherm" (Btu's in 100's) for individual space requirements, radiator capacities, etc.; the "therm" (Btu's in 1,000's) for cast iron boiler sizes; and the "kilotherm" (Btu's in 1,000,000's) for total heat requirements in large buildings, etc. And all you have to do to convert one to another is move the decimal point! Also, as L. P. Hynes[2] points out, conversion to kilowatts per hour is very simple.

Mr. Lewis has started a discussion which is most timely and to which most engineers have something to say. I hope many will say it. Particularly should the "square foot" be threshed out.

* Manager, boiler and radiator division, Crane, Ltd.
[2] See p. 941, November, 1930, issue.

Summing Up, We Have—

By Samuel R. Lewis*

IT seems reasonable now to hope that if we show a generous spirit and a flexible disposition we might receive some more contributions on this subject. I now suggest a modified schedule.

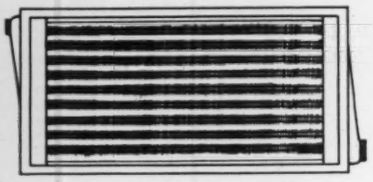
Coined new words very definitely seem to be unpopular. While no one has condemned the new glossary as a whole, someone took a crack at one or the other of the proposed changes until very few of them escaped unscathed.

Here is a new discussion. Do you like the new words and definitions better? If all will enter into this work it will aid improved nomenclature materially.

Horsepower: Dobbin has a hard time passing away.

* Consulting engineer, Chicago, Ill.

There is objection to the exclusive use of the term "watts" by the fan manufacturers because, unless the fan and electric motor are built as a unit by the same manufacturer, confusion might result between watts input and watts output by the motor. Apparently we



A Fancon

could rate a fan by watts output of the motor which runs the fan, but "watts output" requires explanation. Perhaps the horse must remain in the language because of his long established residence, but certainly he should remain only as a small power term, and should be dislodged when he tries to appear as a heat term, in describing the output of a boiler.

A boiler horsepower may mean some arbitrary number of square feet of heating surface anywhere from, say, four to twenty, or may mean ability to evaporate 34½ pounds of water from and at 212 F.

Employment of the term "horsepower," at least as applied to anything but 33,000 foot-pounds per minute, should be deprecated.

Any engineer can become used to thinking and speaking of boiler capacity in pounds of steam evaporated per hour rather than in horsepower.

Kal: This means the steam which ensues when we evaporate 1,000 lb. of water. We do need an accredited term meaning a larger unit than a pound in describing a quantity of steam.

Until the metric system is adopted it would be helpful, at least, to work in decimals, and some of us have expressed ourselves in "M pounds of steam" when we wrote reports and when we meant a "thousand pounds of steam." I suggest, therefore, that while a measure of respectability may well be donated to the terms "C pounds" meaning 100 lb. and "M pounds" meaning 1000 lb., that we adopt and popularize the *Kal* of the *National District Heating Association.*

Unit heater: There is much objection to "elecon." Very well, let us all agree to call it a unit heater, and let that mean a recirculating, fan-equipped comparatively small heating device usually installed without distributing ducts.

Unit ventilator: This is a unit heater which takes part or all of its air from out of doors and which has means for varying the temperature of its air delivery.

Flue convector: "Fluecon" is not approved, but there is reasonable agreement that "concealed radiator" or "encased heater" or "cabinet convector" are unsatisfactory. It *is* a convector and in order to work it *must* go in a flue of some sort. I suggest a compromise and nominate now the uncontracted term, *flue convector.*

Fan convector: I hate to give up "fancon" but fan convector would be consistent with flue convector and it seems a little more satisfactory than blast convector, the alternative suggested by several commentators.

Flue head: "Draft head" is objected to because perhaps some one will feel the draft. "Flue height" is objected to because it might lead to misunderstanding as to what one shall consider the limits in designating the height of a flue. The optimist will be calling it the distance from the bottom of the inlet grille to the top of the outlet grille.

"Flue head" however, will compel the inquirer to look up the matter and should clear the issue definitely as meaning the distance from the top of the flue convector to the bottom of the outlet grille. "Heat head" was suggested, but it does not nail the issue down definitely to the flue, as seems important.

Thermotrap rides along practically unscarred.

Direct vent system is our old friend, the single pipe system, and meets but little criticism. I think the term deserves to live.

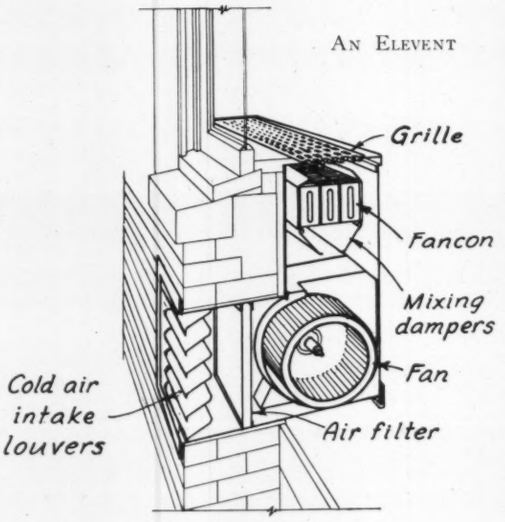
Vacuum system: There is some criticism of a definition for this which makes it mean a pump job, and I admit of course to the purists, that one can obtain a vacuum without a pump; but as nothing in this world is perfect why not let the custom be extended of meaning a pump job when we say "vacuum system?"

Vapor system: The point is well taken by several critics, that the suggested definition does not leave any place in the sun for the two-pipe gravity return which uses a restriction at the inlet rather than at the outlet of the heat transfer agent. The following new definition is suggested:

Vapor system—any steam heating system which uses separate supply and return pipes and which returns its condensation by gravity at atmospheric or less than atmospheric pressure. A "direct vent system" with air valves which do not permit re-entry of air would be called a "direct vent vapor system."

Heat unit: Every one knows that the standard Amer-

An Elevent



ican unit for measuring heat is the British thermal unit. I suggest that we call it a heat unit and drop the "British" and the "thermal."

If a fellow were to print in his catalog that his new furnace was so efficient that it extracted nearly all the pounds out of a ton of coal we would think his copy writer was at least a little peculiar.

Yet there is still much advertising extant which states that the new furnace saves or extracts or transmits so and so much of the heat units, letting no more than neces-

sary of these elusive entities get away. Let it be remembered that a heat unit is not a personality, not an entity, but merely a measure, like a pound or a meter, or a yard. The furnace may conserve the heat, but not the heat unit.

Therm: In our calculations for heat transmission we find the heat unit rather too refined. In any major formula H has so many ciphers that some of them tend to get lost. It is as though we had to describe the Empire State Building as more than 15,000 inches high. The term "therm," meaning 1,000 heat units, has been suggested and is in current practice. I recommend that the use of the "therm" be extended.

The balance of the suggested new usages or limitations were so obvious that very little criticism was made. The substitution of the term "exchanger" for "heater" when describing the heat transfer surfaces in warming water is a happy one. The term "heater" assuredly has been overworked.

Effective Capacity of a Radiator

The day approaches when we shall have a new baby to name; the comfort-efficiency or effective capacity of a radiator or of a flue convector. As far as can today be forecast, eventually our scientists will agree on certain very well circumscribed rating laboratories for these heat transfer surfaces. A standard temperature to be maintained at some standard point within the standard room by a standard radiator the required surface for which in a room of ordinary construction any one can compute, will be agreed upon. The heat input to the standard radiator can be called 100 in the unnamed scale.

If I wish to rate a new heat transfer agent I send it to one of these laboratories and there a constituted authority quickly and economically can determine whether my heater when producing equal temperature at the standard location in the test room under identical conditions to those of the standard radiator, will require more or less heat input than does the standard radiator. My new heater may come back, then, as a "blank 90 per cent" or a "blank 120 per cent" heater of 120,000 Btu input, the "120,000 Btu" being the familiar condensation rating, or the input wattage of an electric heater.

"Blank" is the unnamed child, the comfort rating. I nominated the term "Kent" for the blank term some years ago, after William Kent, the well loved author of Kent's Handbook.

A Diagram for Gas Flow in Pipes

By Leonard L. Hohl

IT IS NOW generally recognized that the most reliable method available to the engineer for investigating the flow of fluids in pipe lines or ducts for the purpose of determining pressure drop is by means of the Fanning formula in the form $2fl\rho U^2/gd$ using for f a friction factor determined by the Reynold's number DUs/Z. The method may be used over a wide range of conditions with accurate results where older formulas could be applied only to a limited range because of the use of constant or slightly varying friction factors intended for special conditions; *e. g.* steam at ordinary working pressures, air at high pressure, air at low pressure, illuminating gas, etc. A recent compilation of gas viscosity data by J. C. Reed and E. E. Ambrosius in HEATING, PIPING AND AIR CONDITIONING[1] has helped to extend the use of this method, especially in the field of refrigerants.

The purpose of this discussion is to illustrate the use of this method as applied to gases by means of examples and to further simplify the solution of such problems by means of a chart. The notation and method are the same as given by Walker, Lewis and McAdams in *Principles of Chemical Engineering.*

f—Friction factor.
L—Effective length of pipe; i. e., straight pipe plus allowance for ells and tees.
ρ—Fluid density in lbs. per cubic foot.
U—Mean velocity in feet per second.
g—32.2 feet per second per second.
d—Actual inside diameter in feet.
D—Actual inside diameter in inches.
s—Specific gravity (water 1), $\rho/62.3$.
Z—Absolute viscosity in centipoises (viscosity relative to water at 68 F.).
$\Delta\rho$—Pressure drop in lb. per sq. ft. (worked out to lb. per sq. in. per 100 ft. in diagram).

[1] Viscosity of Refrigerants, by John C. Reed and Edgar E. Ambrosius, Vol. II, No. 6, p. 455 (June, 1930).

Applying the Viscosity Data to a Steam Flow Problem

Example 1: It is required to find the pressure drop in 100 ft. of 8-in. line (internal diameter is 7.98 in.) with 45,000 lb. of steam flowing per hour. The steam is at 200 lb. pressure absolute and 507 F, corresponding to 125 F superheat.

In this example, D is 7.98 in. and d is 0.665 ft. The volume of the steam is 2.78 cu. ft. per lb. (from steam tables). Specific gravity, s, is 0.359/62.3 or 0.0059. (Fluid density, ρ, divided by weight of 1 cubic foot of water.)

The area of the pipe in square feet is 0.35.

$45{,}000/3600 = 12.5$ lb. steam flowing per second
$12.5 \times 2.78 = 34.8$ cu. ft. steam flowing per second
Therefore, velocity $= 34.8/.35 = 100$ ft./sec.
From Fig. 2, the viscosity in centipoises at 507 F. is $Z = 0.0198$.
Therefore, $s/Z = 0.0059/0.0198 = 0.298$.
Reynolds' number (DUs/Z) is therefore:

$$7.98 \times 100 \times 0.298 = 238$$

From the friction factor portion of Fig. 1, f is found to be 0.0039.
The pressure drop is given by the formula:

$$\Delta\rho = \frac{2fL\rho U^2}{gd} = \frac{2 \times 0.0039 \times 100 \times 0.359 \times (100)^2}{32.2 \times 0.665}$$

$= 131$ lb. per sq. ft. $= 0.9+$ lb. per sq. in. per 100 ft.

A dotted line (marked 1) on Fig. 1 shows the procedure to follow after the volume per second, s/Z, ρ and U have been determined.

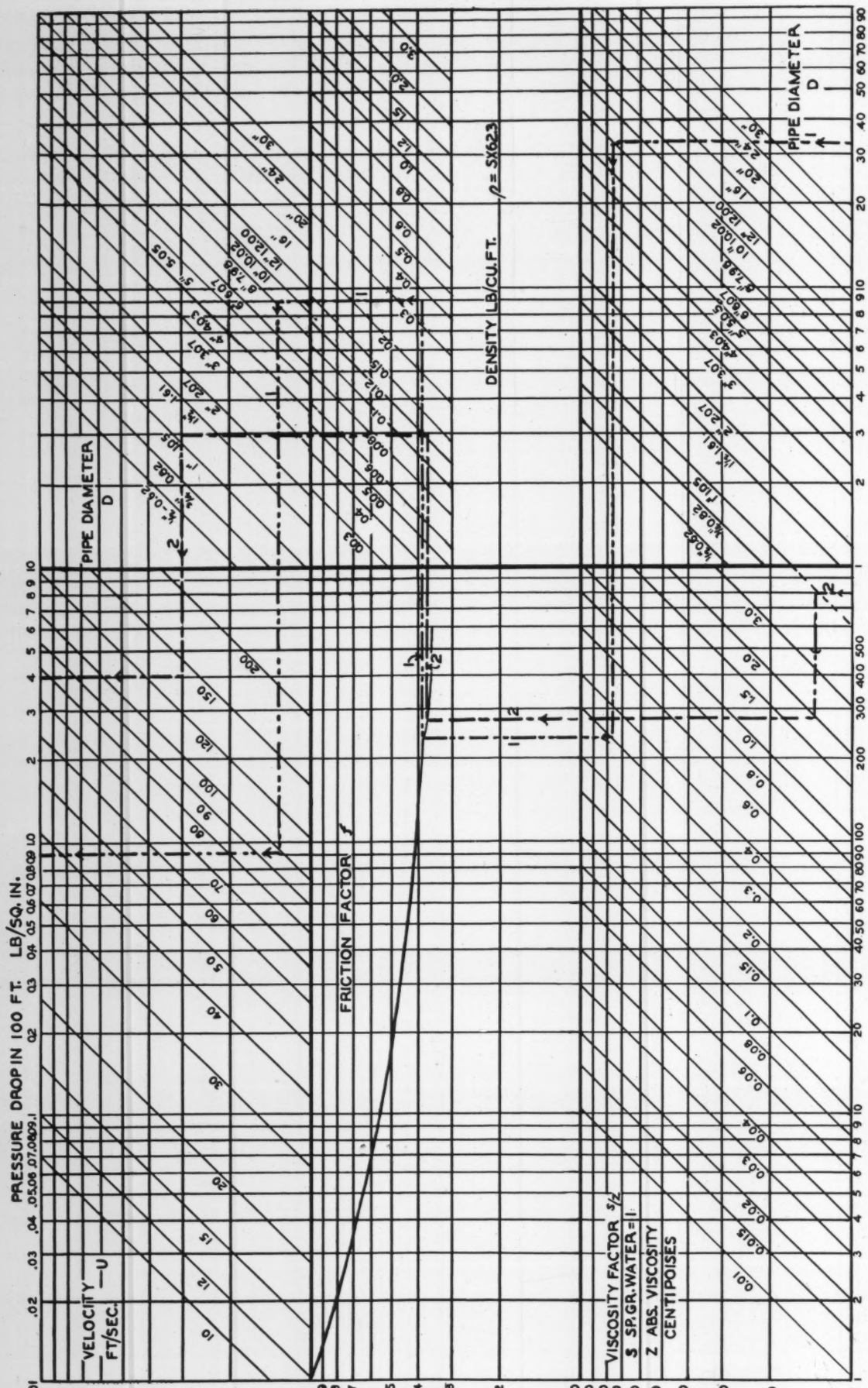


FIG. 1—A CHART FOR COMPUTING THE PRESSURE DROP WHEN A GAS FLOWS IN A PIPE. THE DOTTED LINES INDICATE THE PROCEDURE FOR TWO TYPICAL CASES, WHICH ARE EXPLAINED IN THE TEXT

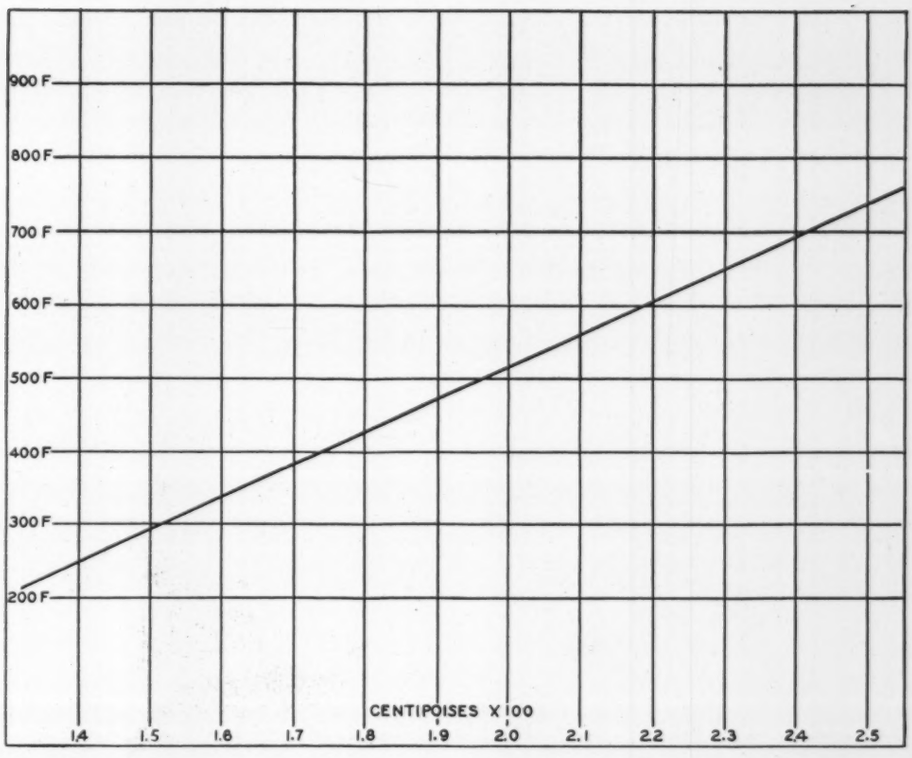


FIG. 2 — VISCOSITY OF STEAM IN CENTIPOISES X 100

How to Use the Chart

The chart (Fig. 1) is entered in the lower right hand corner, using the value of cubic feet per second determined above to locate the point. A vertical line is drawn to the proper pipe diameter line (8 in. in this example). Then a line is drawn horizontally into the viscosity factor portion of the chart to the diagonal line showing $s/Z = 0.298$ (from the value calculated before). The line then goes vertically to the curved line giving the friction factor, f. From there the lines goes to the right, horizontally, to density $\rho = 0.359$. Then the line proceeds upward to the pipe diameter section in the upper right hand corner, turning left at the 8-in. line. It continues horizontally to the velocity portion of the diagram in the upper left hand corner of the chart to $U = 100$. (Because the velocity was determined above to be 100 ft. per sec.). At that point, the pressure drop is determined to be 0.9 + lb. per sq. in. per 100 ft. of pipe, which checks with the value determined in the formula in the preceding section of this article.

Pressure Drop with Butane Flowing

Example 2: Butane (C_4H_{10}; molecular weight, 58) under an average pressure of ten atmospheres (134 lb. gage) at a temperature of 210 F passes through 1-in. standard pipe at the rate of 0.8 cu. ft. per sec. Find the pressure drop per 100 ft. of pipe.

In this example, D is 1.05 in. and d is 0.0875 ft. The area of the pipe is 0.006 sq. ft. The weight per cubic foot, ρ, is 1.19 and s is therefore 1.19/62.3 or 0.019. The absolute viscosity, Z, is 0.01 centipoises. (See *Viscosity of Refrigerants*[1].)

Therefore, $s/Z = 0.019/0.01 = 1.9$
Velocity, $U = 0.8/0.006 = 133$ ft. per sec.
Reynolds number $(DUs/Z) = 1.05 \times 133 \times 1.9 = 266$.
The friction factor (from Fig. 2) corresponding to 266 is $f = 0.0039$.
The pressure drop is therefore:

$$\frac{2fL\rho U^2}{gd} = \frac{2 \times 0.0039 \times 100 \times 1.19 \times (133)^2}{32.2 \times 0.0875}$$

$= 5830$ lb. per sq. ft. $= 40$ lb. per sq. in.

In following this example through the chart shown as Fig. 1 it will be noted that the line for 1-in. pipe must be extended to reach the 0.8 cu. ft. per sec. line as shown by the dotted line used. Similar extensions may be made in the first, second, fifth and sixth steps of the diagram if necessary. It will also be noted that the density line for 0.12 is used (1/10th of the actual value) to avoid running off the next diagram. This leads to 4 lb. pressure drop which must be multiplied by 10 to offset the 1/10th density used in the density diagram. (The example with butane shows an excessive pressure drop but was selected to illustrate two possibilities of extending parts of the diagram.)

It will be noted that in both the above examples the friction factor is 0.0039 (roughly 0.004). This factor, 0.004, may be used as a rough average value for most cases of gas flow in pipes in which a high degree of accuracy is not required and thus reduce the labor of working such problems. If this approximate method is chosen it is only necessary to enter the diagram at the fourth step (density in lb/ft.^{3}) ignoring the first three sections; and proceed from the intersection of the horizontal 0.004 line and the density line up into the pipe diameter section across into the velocity section thence up to the final result. For this abbreviated approximate method it is necessary to determine the density and velocity before entering the diagram.

The shortened method may also be used for gas flow in which no viscosity data is available for the gas in question. Where reasonably accurate results are required and viscosity data are available the full procedure is recommended. This latter procedure requires in addition to the density (lb/ft.^{3}) and velocity (feet per second), three more quantities, cubic feet per second and specific gravity (density/62.3) and viscosity in centipoises before proceeding as previously outlined.

The friction factors in the diagram are the values calculated by McAdams.

Value of Using Consistent Units Throughout

The method presented by Mr. Hohl (in his article on p. 393) for the determination of the frictional pressure loss in gases flowing in pipes offers a convenient and rational scheme to the engineer. However, while this method eliminates much of the "rule-of-thumb" in regard to the selection of the friction factor to be employed in the Fanning formula, there are a number of other items which must be given consideration when applying it. A large pressure drop in proportion to the final absolute pressure, or a large temperature change in the gas due to external conditions, may alter the density and viscosity so as to materially affect the result.

More Experimental Work Needed on Large Pipe

While the curve in Fig. 1 is the result of a compilation of a number of investigations which covered a large range in pipe sizes, it is felt that more experimental work is needed on large diameter steel and cast iron pipes to definitely prove that one curve will suffice for all sizes. Possibly modern pipe of large size is relatively less rough internally than the smaller pipe, in which case the curve for the larger pipe should approximate more closely the one deduced from the investigations of Stanton and Pannell, and Saph and Schode, on drawn brass pipe. The latter curve has values of the friction factor approximately 15 per cent lower than those for cast iron pipe. However, use of the diagram presented here will give results on the safe side.

Use of Consistent Units

A few remarks on this general theory of fluid flow might be apropos. It has been established that a definite relation (such as the curve shown in Fig. 1) exists between the friction factor f and Reynolds' number for pipe of a certain degree of roughness. Since both of these functions are dimensionless ratios it follows that if all of the variables which enter into these two ratios, such as velocity, density, viscosity, etc., are expressed in *any consistent set of fundamental units* the corresponding numerical values of f and Reynolds' number will always be the same.

For example, if the fundamental units which are mass, length, and time are expressed as pounds, feet, and seconds in *all* of the variables involved the resulting numerical values of f and the corresponding Reynolds' number will be identical with those arrived at if *all* the variables had been expressed in grams, centimeters, and seconds.

Thus, the work of one investigator using the English units is immediately comparable to the results of another employing the centimeter-gram-second system. To illustrate the advantage to be gained by adhering to *any* consistent set of units, the change from viscous flow to turbulent flow always occurs at a value of Reynolds' number between 2,000 and 2,500. The simplicity of application and lessening of the possibility of error when using formulas and charts containing all English or all metric units will be appreciated by the practicing engineer. Why not express absolute viscosity values as pounds per foot per second instead of centipoises?—*R. E. Gould.*

Some Limitations of the Chart

In studying Mr. Hohl's paper, proposing a diagram for finding the pressure drops in pipes carrying gases, the writers find this method fairly satisfactory. The chart did, in several trials, check consistently low by several per cent, as compared with the Fanning formula calculations given by Mr. Hohl.

This, in a measure, we feel is due to the fact that the chart is rather small. Charts of this kind which are to be used frequently should be made as large as conditions will permit in order to secure maximum accuracy. This will insure that maximum accuracy will be obtained at all parts of the chart and it should be constructed so that the per cent accuracy will be the same in each portion.

Such a diagram as Mr. Hohl proposes, plotted to a suitable large scale, would be quite useful to one having many routine problems to solve along this line. However, for one having only a few cases to consider it would possibly be better to make the actual calculations.

We feel that Mr. Hohl could have used a better choice of units in his Reynolds' number and formula for calculating pressure drops. In his system, the Reynolds' number is not a dimensionless quantity. The same reasoning applies to the friction factor. The author should have calculated his friction factor, Reynolds' number on a strict dimensionless basis, as it would then have been more useful. As it is, such factors as specific gravity

just help to complicate matters and touch upon the accuracy of the chart.

This chart we feel could be much improved upon if it were made up with a consistent set of units. This also applies to the temperature-viscosity curve. The English system, no doubt, would be the better system, since it is the foot, pound, second system used by the engineer, who would be interested in such things as finding the pressure drops in pipes where we have gas flow.

In predicting pressure drops where we have steam or other heated gases flowing and have only the initial conditions available, one must be careful; otherwise serious errors may result from our conclusions. For instance, this chart is based on 100 ft. of pipe length and the pressure drops are also based on that length. When the drop is found for 100 ft. and is wanted for 1,000 ft., it will not necessarily be ten times what it was for 100 ft. The drop for the last hundred feet will be different from the drop in the first hundred feet of the thousand foot run. These facts are especially pronounced when a very high velocity exists, where we have a high pressure drop with a relatively low temperature drop. However, when a very low velocity exists, the pressure drop will be very low and the temperature drop will be very high. All these factors will have a direct bearing on the pressure loss and yet will not be shown by the chart method of calculation. We, therefore, suggest that this chart only be used for pipe lengths up to possibly not much more than 200 ft. Also, it should not be used where the pressure drops are more than 15 per cent, and especial care should be exercised when used where the temperature drop becomes very large.

We doubt very much that the second example as given by Mr. Hohl will work out as predicted under actual conditions. It is a general assumption that the Fanning formula will hold only when the pressure drop is as low as 10 or possibly 15 per cent, and that it can be made to work when the pressure drop gets as high as 40 or 45 per cent, when average conditions of pressure and temperature are assumed in the pipe. Of course, under such conditions we must know the final pressure, and then there is no need for calculating it. Basing the calculations on original data only, the Fanning formula will not hold when the drop becomes as large as proposed by Mr. Hohl in his second example. Of course, we realize that is just an example used to illustrate the use of the chart and would not exist in actual practice.

Where such excessive pressure drops exist, the Fanning formula will have to be modified to take care of the expansibility of the gas.—*John C. Reed, Edgar E. Ambrosius.*

The Author's Reply

The comment concerning consistent units has, of course, some merit, but the writer had in mind, when choosing the units of the article, to adopt those in common use among engineers—especially those of the petroleum and gas industry. As pointed out by Messrs. Reed and Ambrosius as well as Mr. Gould, where great pressure drops occur, the method becomes inexact because of expansion of gases. In many such cases a pipe line may be divided into sections each of which will not have more than 10 to 15 per cent drop.

As mentioned by Messrs. Reed and Ambrosius this or similar diagrams are useful where a considerable number of problems must be worked out, and in these ordinary cases charts have saved time as well as aiding in checking work in which a greater degree of accuracy than that of a chart is required.

In conclusion the writer wishes to point out the fact that the velocity is rarely below the critical and that the friction factor does not vary over a wide range in ordinary gas flow problems.—*Leonard L. Hohl.*

Steam *from Contented* Reducing Valves

By Henry G. Schaefer

ONE part of piping installations susceptible to error in design, and consequently provocative of profanity when such error manifests itself in the form of weird and ungodly noises in operation, followed by wire-drawn seats screeching for the attention of the harrassed operating engineer, is the pressure reducing valve.

Why is it that a peaceable, law-abiding designing engineer, whose savage, sub-conscious instincts are well tramelled by years of the 8:08 and 5:17 cycle, will invariably revert to primitive type upon the occasion of his being confronted by the problem of sizing a reducing valve?

An Example

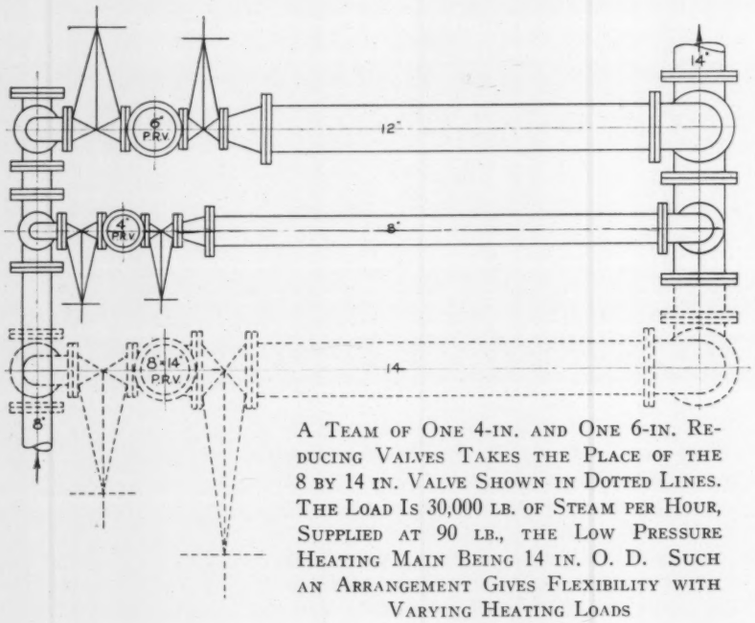
Here is a sample of the process: The maximum computed heating load is 30,000 lb. of steam per hour. Steam is available at 90 lb. pressure. The low pressure main is 14 in. O. D. A misdirected, though creditable, effort toward economy, and the valve manufacturer's catalog, combine to produce the conception of an 8 in. x 14 in. enlarged outlet reducing valve. Presto—the job is done—that is until the chief engineer starts his daily calls (which, by the way are the direct cause of the telephone company's removing his 'phone and refusing him further service).

In piping engineering, as in every other branch of human activity, all goes well until man gets the idea that he knows more than Mother Nature, and starts to highhat her. Then the fun begins. That estimable dame is a great believer in dividing responsibility. She does not expect us to run with one leg, nor to pack a valve with one hand. She gave us two ears to be sure we would not miss the five o'clock whistle, and a pair of eyes to watch the gage. Yet, in the face of all these examples, most of us will persist in sending a reducing valve out

into the world alone. It isn't right—not technically, not biologically, nor yet economically.

The fact of the matter is that in nine-tenths of all heating installations the maximum estimated load is never reached in practice. When this same designing engineer is called upon for an estimate of the fuel cost for the year does he begin by multiplying the 30,000 lb. by the number of hours in the heating season? No siree! *You* know what he does.

To rough the situation out, during 80 per cent of the operating time the load is less than 40 per cent of the maximum. It follows that in many installations a division of the work between two valves, one of twice the capacity of the other, would be desirable, the little fellow working most of the time and the big chap taking over the task at the moderate peaks—and certainly that is natural enough, for I guess we are all agreed that the little fellows do most of the world's work. Then of course, when the mercury disappears altogether, they both put their shoulders to the wheel to survive the depression.



A Team of One 4-in. and One 6-in. Reducing Valves Takes the Place of the 8 by 14 in. Valve Shown in Dotted Lines. The Load Is 30,000 lb. of Steam per Hour, Supplied at 90 lb., the Low Pressure Heating Main Being 14 in. O. D. Such an Arrangement Gives Flexibility with Varying Heating Loads

In the case mentioned above a team consisting of one 4-in. and one 6-in. would just about do the job perfectly, each having the same inlet and outlet size. Why not an enlarged outlet type? Well, we'll come to that later.

Just now let us hook up all three in parallel, each with a control gate valve fore and aft, and see what happens as we cut in the 8-in. x 14-in. on a mild day. There goes the white elephant on the 20 per cent load, loafing along with an attempt at nonchalance, but with spindle jiggling up and down, diaphragm with St. Vitus' dance, gradually pitting disc chattering on a seat that is already showing signs of wire-drawing, and every little while showing an unmistakable leaning toward building up to a point that might cause damage to the delicate bellows of the vacuum traps—in short, showing all the signs of an impending nervous breakdown. And what a racket!

Well, that is that. Let's take her out and cut in the 4-in. See how gracefully it takes the light load. Here is the job that valve was built for, and it takes to the task like a duck to water.

In the morning the mercury is down to 26 degrees. The 4-in. valve is wide open and showing signs of getting noisy. Stand by while we cut in the 6-in. and give the little fellow a rest. Notice how easily and confidently the big brother steps in—just like the youngster tackled the light load yesterday.

Sizing the 4- and 6-in. Team

And now about the straight sizing of the 4-in. and 6-in. team. What causes all that hissing in a reducing valve? Why, the speed at which the gas is traveling, and then it is only natural that the noise is increased when, in addition to its velocity, its flow is broken up in the whirls and eddies caused by its passage through the body of the valve.

Not a great amount of imaginative effort is required to gain a conception of the wonderful pipe-organ effect produced when, under a light load, the hissing 90-lb. steam squirms through the slit between the seat and disc of that 8-in. x 14-in. giant, irksomely trying to do a boy's work, and enters the 4- or 5-lb. zone in the capacious maw of that leviathan.

You will notice that, in piping the 4-in. and 6-in. team, a gate valve is bolted directly to the outlet flange, and that to the valves there are bolted 4-in. x 8-in. and 6-in. x 12-in. tapered increasers, respectively, followed by ten feet (or ten pipe diameters) of straight pipe. On account of the comparatively restricted space available to the steam from the reducing valve seat to the start of the flare of the increaser, much of the actual reduction in pressure, or, what amounts to the same thing, the expansion of the gas, occurs in the smoothly flared fitting, and not in the tortuous channels of the valve body.

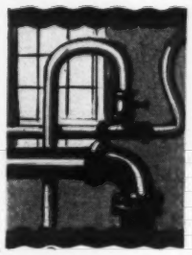
Arrangement Offers Gradual Expansion

Since the velocity of a given weight of steam through a fixed channel in a unit of time at 90 lb. pressure is only a fifth of that of the same weight under a pressure of 3 pounds, it is apparent that the arrangement described offers gradual expansion in successive steps. And then, that ten feet of straight pipe is very important in the quieting treatment in that it provides a convalescing chamber for our "water gone crazy with the heat."

The fact that all these courtesies are gratefully appreciated is amply proved by a total lack of those awful yowls of anguish that emanate from an improperly sized and located reducing valve and which are replaced by a contented purr.

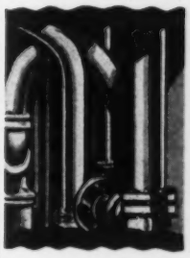
Like other products of the milk of human kindness, the reducing valve cheese is particularly delectable, in that the saving in time, annoyance and cost in maintenance is unaccompanied by an additional initial investment bill, but rather, in most cases, by an actual saving.

The only cost is the investment of a full half hour spent in a study of the actual body (not a catalog cut) of the reducing valve to be used. The case of the heating load considered is typical. Practically all reducing valve duties may be similarly analyzed, and then treated as the analysis dictates.



Because of the general interest recently shown in the question of the bending of curved pipes during flexure, Sabin Crocker offers in this article a brief explanation of the theory and points out its practical applications.

Drawing freely on the work of Hovgaard, Karman, Timoshenko, Wahl and others, Mr. Crocker's paper offers a comprehensive resumé, which should be of considerable interest to the designer of piping systems.



Bending of Curved Pipes

By Sabin Crocker*

A PAPER entitled "Bending of Curved Pipes" was presented by Professor William Hovgaard [1] at the *Third International Congress of Applied Mechanics* held at Stockholm, Sweden, last summer. This paper was a noteworthy summary of four papers by the same author published in the *Journal of Mathematics and Physics* of the Massachusetts Institute of Technology [2] and a fifth paper published in the *Proceedings* of the National Academy of Sciences.[3]

The theories advanced by Professor Hovgaard regarding flattening of the circular cross section of curved pipes during flexure are supported by a comprehensive series of tests and substantiated by the work of Karman,[4] Timoshenko [5] and Wahl.[6] The formulas presented by these authors are in essential agreement, although the symbols and methods of derivation used are somewhat different. Their results taken together offer conclusive proof that flattening of the circular cross section of curved pipes during flexure must be taken into account in an accurate flexibility analysis. In view of the general interest recently manifest regarding this subject, it will not be amiss to offer a brief explanation of the flattening theory to the readers of this magazine.

Flattening Theory

The flattening theory takes into account the fact that curved pipes behave differently during flexure than do straight pipes or solid curved bars. A curved pipe having a wall which is comparatively thin with reference to its diameter tends to flatten during flexure instead of having its cross section remain circular as is the case with a straight pipe or solid curved bar. The amount of flattening in a curved pipe is determined by the diameter of the pipe, its wall thickness, and the radius to which it is bent, as explained later. The actual characteristics of curved pipes have been found to check the flattening theory in a large number of load deflection tests supported by observations with strain gages.

The theory of flattening of the circular cross section of curved pipes during flexure applies to pipes which are initially curved, such as quarter bends, U-bends and the like. Straight pipe obeys the ordinary laws of flexure without this extra complication, neglecting, of course, the relatively small amount of curvature thrown into straight pipe through bending moments in the line.

The theory of flattening during flexure as ordinarily applied does not take into account any initial flattening of the circular cross section of a bend which may have been produced in manufacture. The amount of initial flattening in carefully made bends should not exceed one to two per cent of the pipe diameter and usually is neglected in the computations. Cases have been reported, however, where the ellipticity ran as high as 5 to 10 per cent, but it is believed that this represents the exception rather than the rule. A method of computing the change in bursting stresses due to initial ellipticity developed by Wahl, Bowley and Back [7] is referred to later in this article, where it is shown that these stresses tend to neutralize the transverse bending stresses set up by flattening during flexure.

From a practical standpoint, it is inadvisable to try to take into account all points of this kind which cannot be accurately estimated in advance, since in addition to ellipticity due to flattening in fabrication, commercial tolerances in pipe manufacture allow: (a) a plus or minus wall thickness variation of 12½ per cent which in extreme cases may change the moment of inertia something in the order of ± 10 per cent; (b) one per cent plus or minus out-of-round on the diameter. Usual practice is to base flexibility computations made for de-

*Engineer, the Detroit Edison Company, Detroit, Mich.

[1] Professor of Naval Architecture, Massachusetts Institute of Technology.

[2] "The Elastic Deformation of Pipe Bends," Vol. 6, No. 2, Nov., 1926; "Deformation of Plane Pipes," Vol. 7, No. 3, October, 1928; "Further Research on Pipe Bends," Vol. 7, No. 4, Dec., 1928; "Tests of High Pressure Pipe Bends," Vol. 8, No. 4, 1929; all by Professor William Hovgaard and published in the *Journal of Mathematics and Physics* of the Massachusetts Institute of Technology. These papers also have been reprinted by that institution and are available in pamphlet form.

[3] "Bending of Curved Tubes," by William Hovgaard, *Proceedings* of the National Academy of Sciences, Vol. 16, No. 6, June, 1930, p. 444.

[4] "Regarding the Change in Shape of Thin Walled Tubes, Especially Thin Walled Pipe Bends," by Th. Karman, *Zeitschrift des Vereines Deutscher Ingenieure*, Bd. 55, 1911, p. 1889.

[5] Discussion by Professor S. Timoshenko of paper by Crocker and Sanford on, "Elasticity of Pipe Bends," *A. S. M. E. Trans.*, 1922, p. 585.

[6] "Stresses and Reactions in Expansion Pipe Bends," by A. M. Wahl, *A. S. M. E. Trans.*, 1928, Paper FSP-50-15, p. 241.

[7] "Stresses in Turbine Pipe Bends," by A. M. Wahl, J. W. Bowley and G. Back, *Mechanical Engineering*, Nov., 1929, p. 823.

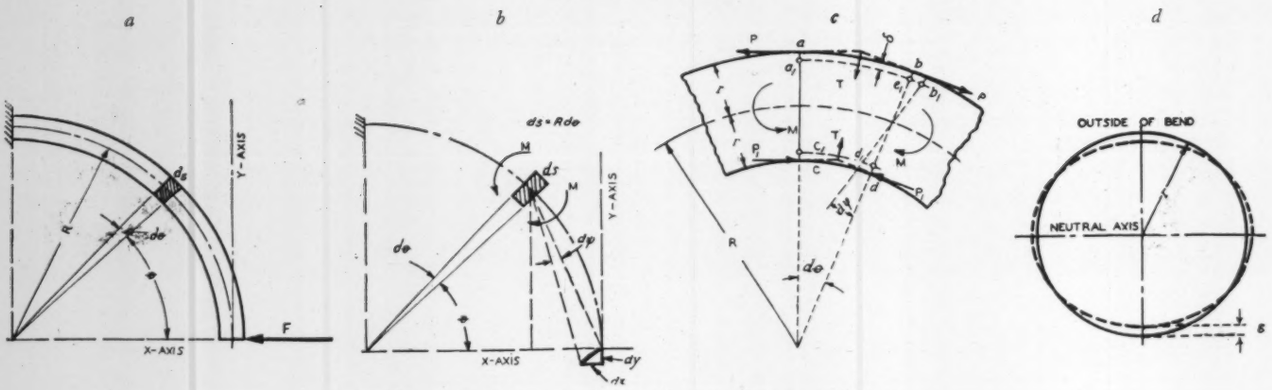


FIG. 1—BENDING OF CURVED PIPE IN PLANE OF BEND

sign purposes on the assumptions that the pipe is of nominal thickness and that it is initially round, and to allow for indeterminate factors by designing for a working stress which is a safe margin below the yield point of the material at the operating temperature.

Briefly stated, flattening of curved pipes during flexure comes from the fact that tensile and compressive stresses induced by flexure cannot act in a straight line and consequently must have components toward the neutral axis of the cross section. This situation is illustrated in Fig. 1. At *a* is shown a conventional quarter bend fixed at one end and with a simple force F applied at the other. An elementary section ds is considered by itself at *b*, where it is seen that the opposite ends of ds are acted on by a bending moment M induced by the force F. An enlarged view of ds is shown at *c* with the tensile and compressive stresses induced by bending indicated as P-P and P'-P'.

Fig. 1*c* shows that tensile and compressive stresses in the longitudinal fibers of the bend cannot act in a straight line and hence have components T acting toward the neutral axis of the cross section. Since no corresponding sidewise components act on the pipe wall, there is a tendency for the original circular cross section to flatten out as shown by the dash lines in Fig. 1*d*.

Effect on Stress Distribution

Transverse (hoop) bending stresses are set up by the flattening effect described above, and in addition there is a re-distribution of the longitudinal bending stresses. The derivation of formulas for computing these stresses involves more mathematics than can be introduced here and it is suggested that those interested refer to the papers mentioned at the beginning of this article, or to the "Piping Handbook," by Walker and Crocker. A typical illustration of the redistribution of longitudinal bending stress is shown in Fig. 2, while the corresponding transverse stresses set up are shown in Fig. 3.

The ordinary formula for longitudinal stress due to bending is $S = My/I$, where S is stress in lb. per sq. in., M bending moment in in.-lb., y the distance of the element from the neutral axis in inches, and I the moment of inertia of the section in inches to the fourth power. The modified formula for longitudinal bending stress in the outside wall according to the theory for flattening during flexure is:

$$S_l = \frac{My}{KI}\left[1 - \frac{6y^2}{r_o^2(5+6h^2)}\right] \qquad (1)$$

where,

r_o is the outside radius of the pipe in inches, and h and K are as defined below under "Rigidity Multiplication Factor K." To obtain the stress at any intermediate point in the pipe wall, substitute the corresponding radius in place of r_o.

The transverse stress at any point in the outside of the pipe wall, according to the theory for flattening during flexure is:

$$S_t = \frac{18\,Mr_oh\left(1 - \frac{2y^2}{r_o^2}\right)}{I\,(1+12h^2)} \qquad (2)$$

This stress is a maximum at the points where $y = r_o$ and $y = 0$ as shown in Fig. 3. At these points the expression reduces to

$$S_t = \pm\frac{18h}{1+12h^2} \times \frac{Mr_o}{I} \qquad (3)$$

The principal effect of initial flattening of a bend during its manufacture is to alter the distribution and magnitude of the transverse (hoop) bursting stress due to internal pressure. Instead of approximating uniform

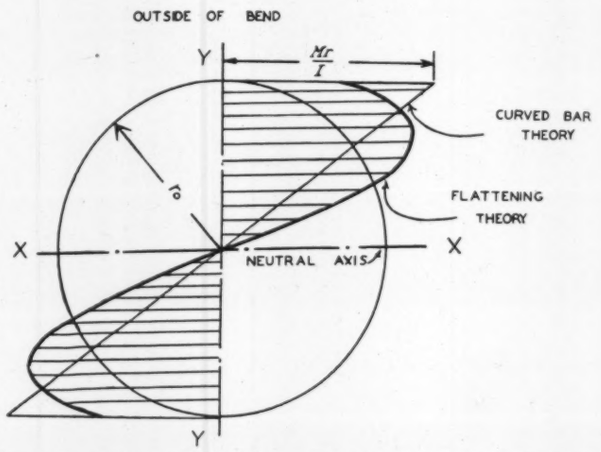


FIG. 2—LONGITUDINAL STRESS DISTRIBUTION IN OUTSIDE WALL OF CURVED PIPE. $h = tR/r^2 = 0.8$. TENSILE STRESS SHOWN TO RIGHT OF Y AXIS, COMPRESSIVE TO LEFT

tension across the pipe wall and around the cross section as would normally be the case, the bursting stress becomes distorted as shown in Fig. 4. According to Messrs. Wahl, Bowley and Back[7] the stress at the maximum points is located in the outer wall of the

pipe at the inside and outside of the bend and can be computed from the relation $S' = 6M_o/t^2$, where $M_o = pr_i\delta$ = bending moment in inch lb. per inch of axial length at point of maximum stress; p = internal fluid pressure in lb. per sq. in.; r_i = internal radius of pipe in inches; δ = ellipticity in inches. On this stress it is necessary to superimpose the transverse stress due to internal pressure found from the common formula $S_1 = pr_i/t$ as follows:

$$S_t' = S_1 + S'$$

$$S_t' = pr_i/t + 6M_o/t_2 = \frac{pr_i}{t}\left(1 + \frac{6\delta}{t}\right) \quad (4)$$

The approximate distribution of this stress is shown in Fig. 4 for a 10-in. standard weight pipe (0.365 in. wall) having a maximum ellipticity of 0.1 in. and a working pressure of 400 lb. gage. The maximum stress is computed by substituting in *Equation* 4 the values: $p = 400$; $r_i = 5.01$; $t = 0.365$; $\delta = 0.1$

$$S_t' = \frac{400 \times 5.01}{0.365}\left(1 + \frac{6 \times 0.1}{0.365}\right) = 14{,}500 \text{ lb. per sq. in.}$$

Fortunately, the rather high maximum stresses indicated by *Equation* 4 and shown in Fig. 4 are opposite in sign to the maximum transverse bending stresses due to flattening during flexure as shown in Fig. 3.

A comparison of Figs. 3 and 4 indicates that the two stresses tend to neutralize each other quite effectively around the pipe cross section so that for the case considered the combined stress is lower than either stress would be if considered alone. In general the effect of initial flattening is to offset transverse stress induced by flattening during flexure, and from this standpoint is advantageous, if any thing, in curved pipes, especially where the transverse stress due to flexure is high.

This tendency for the transverse stresses to neutralize one another is a further argument in favor of Professor Hovgaard's proposal (explained later) that the design of curved pipes be based on longitudinal stress considerations. This possible advantage of initial flattening with curved pipes does not exist with straight pipes, since they have no transverse bending stress induced by flexure.

Rigidity Multiplication Factor "K"

Due to the fact that the fibers farthest removed from the neutral axis tend to relieve themselves of longitudinal stress through flattening of the circular cross section during flexure, there is a lessening in the resistance offered by the bend to flexure which is equivalent to a reduction in the moment of inertia I of the pipe section. Consequently in dealing with a curved pipe, it is necessary to consider, not the apparent flexural rigidity EI, but a value of flexural rigidity KEI, where K is a factor less than unity (which is called the rigidity multiplication factor) determined from the relation:[8]

$$K = \frac{1 + 12h^2}{10 + 12h^2}, \text{ where } h = \frac{tR}{r^2} \quad (5)$$

where

t = wall thickness of pipe, inches; r = the mean radius of the pipe cross section, inches; and R = the mean radius of the bend, inches. Numerical values of K for various values of h are plotted in Fig. 5.

To give an idea of how h and K vary for actual cases, 10-in. standard weight pipe (0.365 in. wall) bent to a radius of six pipe diameters has an h value of about 0.8 and a corresponding K of 0.5, while a 10-in. 1350 lb. *A. S. A.* pipe (1.06 in. wall) bent to the same radius has an h value of 2.0 and a corresponding K of 0.84. In other words, the flattening theory shows that the 10-in. standard weight pipe bend is twice as flexible as the ordinary curved bar formulas would indicate, and that 1350-lb. pipe is $1 \div 0.84 = 1.2$ times as flexible.

The closer approach of 1350-lb. pipe to obeying curved bar formulas is to be expected from the fact that it is, comparatively speaking, a thick walled tube with a high resistance to flattening. The foregoing remarks regarding relative flexibility apply to the relation of deflection to reacting force as determined from such formulas as

$$\Delta = \frac{0.7854\ FR^3}{KEI} \quad (6)$$

where Δ is deflection in inches in the line of action of force F for a quarter bend loaded as shown in Fig. 1*a*. The more complicated relation of stresses is treated in a later paragraph.

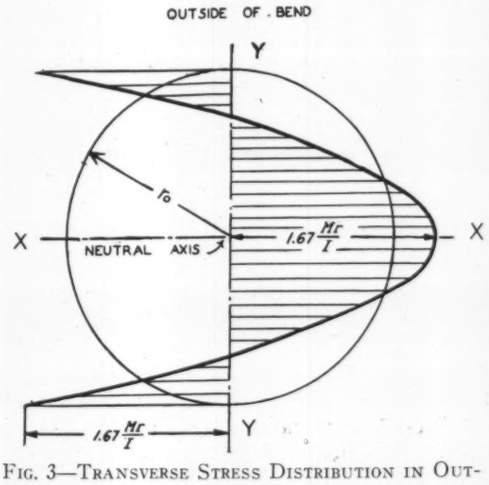


FIG. 3—TRANSVERSE STRESS DISTRIBUTION IN OUTSIDE WALL OF CURVED PIPE. $h = tR/r^2 = 0.8$ TENSILE STRESS SHOWN TO RIGHT OF Y AXIS, COMPRESSIVE TO LEFT

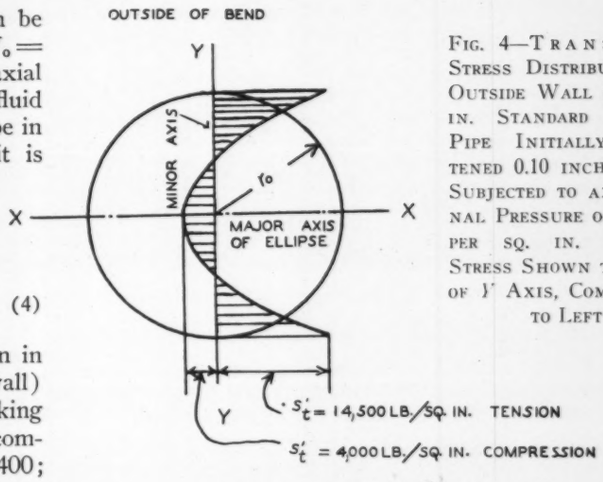


FIG. 4—TRANSVERSE STRESS DISTRIBUTION IN OUTSIDE WALL OF A 10-IN. STANDARD WEIGHT PIPE INITIALLY FLATTENED 0.10 INCH, WHEN SUBJECTED TO AN INTERNAL PRESSURE OF 400 LB. PER SQ. IN. TENSILE STRESS SHOWN TO RIGHT OF Y AXIS, COMPRESSIVE TO LEFT

Safe Stresses for Curved Pipes

In deciding what are safe working stresses to allow in curved pipes, consideration must be given to how the

[8] The K given here is that used by Karman, Timoshenko, Wahl and in the *Piping Handbook*, by Walker and Crocker. It is the reciprocal of that used by Professor Hovgaard, and is preferred by the writer in its present form as being an easily understood coefficient to apply to EI.

various stresses due to internal pressure and expansion will combine. Sometimes, also, it is necessary to take into account bending stresses due to dead weight between supports, but this follows along without extra complication when the general principles are understood. The case discussed first will involve piping in one plane only, which is subject to bending and bursting stresses due to internal pressure, but which is free from torsion.

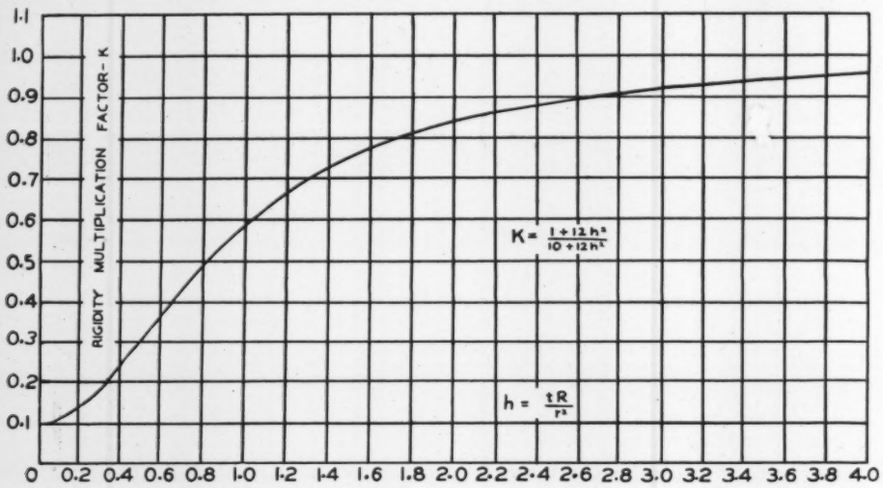


FIG. 5—CURVE FOR DETERMINING RIGIDITY MULTIPLICATION FACTOR K FOR CURVED PIPE

Piping in one plane, free from torsion: According to the maximum shear theory, when a ductile material is subjected to the action of two or three stresses on mutually perpendicular planes, failure is due to a permanent set or sliding in the plane of maximum shear and will occur only when this stress reaches a certain limiting value.[9] The maximum tangential or shear stress is one-half the difference of the maximum and minimum tensile stresses acting on two of the mutually perpendicular planes, neglecting intermediate stress on the third plane. Its limiting value is one-half the stress producing failure in pure tension or compression (compression is treated as negative tension).

For the case of piping in one plane, free from torsion, and subjected to stresses due to internal pressure and to expansion, the stresses in the three mutually perpendicular planes are: (1) longitudinal bursting stress due to internal pressure plus longitudinal bending stress due to expansion; (2) transverse bursting stress plus, in the case of curved pipes, transverse bending stress due to flattening of the circular cross section; (3) direct radial stress perpendicular to the pipe wall due to internal pressure—a compression numerically equal to the fluid pressure at the inside of the pipe wall and zero at the outside.

Usually (1) will be the maximum stress and (3) the minimum stress, leaving (2) as the intermediate stress which is neglected. The equivalent tensile stress to consider in design then becomes twice the maximum shear stress of the maximum shear theory or

$$S_{eq} = S_{max} - S_{min} \qquad (7)$$

noting that compressive stresses are negative tension and reverse the signs. S_r usually is small enough to neglect and the equivalent tensile stress can be considered as the longitudinal stress S_{max} which is the arithmetical sum of the longitudinal bending and pressure stresses. This gives a simple criterion for establishing safe design stresses where (2) above is neither the maximum or minimum stress in the three mutually perpendicular planes.

If (2) is the maximum stress due to a preponderance of hoop tension produced by fluid pressure, it becomes the criterion for design rather than the longitudinal stress, noting that allowable values for bursting pressure are not necessarily the same as for bending stress as explained in a later paragraph. On the other hand (2) may be a large compressive stress (negative tension) due to excessive flattening of the circular cross section (see Fig. 3) and hence represent the minimum stress in the three planes. In this case the equivalent stress becomes

$$S_{eq} = S_l - (-S_t) = S_l + S_t \qquad (8)$$

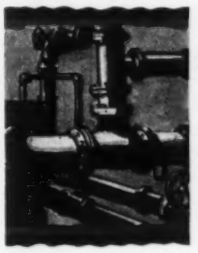
which may exceed the elastic limit of the material, especially when pressure stresses are included with S_l and S_t.

An outstanding feature of Professor Hovgaard's four latest papers was his demonstration that the *maximum longitudinal* stress in the middle fibers of the pipe wall can be used as a safe criterion for bending stresses in curved pipes, disregarding transverse bending stresses incidental to flattening of the circular cross section. If the transverse (compressive) bending stress in the outside wall at the top of a bend is combined with the longitudinal (tensile) stress at that point, excessively high values for the equivalent stress are obtained which apply only in a very localized layer of material at the outer edge of the pipe wall.

This high local stress may be considered as relieved by plastic flow and disregarded in the computations. For ordinary mild steel or wrought iron pipe material having a yield point of not less than 25,000 lb. per sq. in., Professor Hovgaard states that satisfactory results will be obtained provided the *longitudinal* working stress in the middle fibers of the pipe wall does not exceed 16,000 lb. per sq. in. This working stress should be considered as the total combined stress including the longitudinal stress due to internal fluid pressure, minus any direct compression due to expansion thrust at anchor

[9] See *Applied Elasticity*, by Timoshenko and Lessells, First Edition, p. 501, or other recent text on strength of materials.

points. Appreciable permanent set will not occur unless this stress exceeds 20,000 lb. per sq. in. Presumably Professor Hovgaard intended his recommendations to be limited to temperatures not exceeding 750 F, although no mention of operating temperature was made in his papers.



The maximum longitudinal bending stress referred to is that existing in the mid-fibers of the wall at a point corresponding to the point of maximum stress shown in Fig. 2 which can be found by making the proper substitutions in *Equation* 1. Determination of the maximum stress from that formula would be a rather laborious cut-and-try process, however, and Professor Hovgaard has indicated the following simplification for computing this stress directly:

$$S_1 = \beta \frac{Mr}{I} \text{ where } \beta = \frac{2}{3K}\sqrt{\frac{6h^2+5}{18}} \qquad (9)$$

β, which is known as the maximum longitudinal stress multiplication factor, has been plotted against $h = tR/r^2$ in Fig. 6 for convenience in obtaining values of β to use in stress computations.[10]

Pipe lines in space, subject to both bending and torsion: Pipe lines in space usually are subject to torsion as well as to bending and pressure stresses. Where torsion exists, it can be taken into account through the maximum shear theory by combining it with the other stresses in the following equation:

$$S_{eq} = 2\sqrt{S_s^2 + \left(\frac{S_{max} - S_{min}}{2}\right)^2} \qquad (10)$$

where

S_s is shearing stress due to torsion, S_{max} the sum of the longitudinal stresses due to bending and pressure, and S_{min} the direct radial compression due to pressure (which changes the minus sign to plus, but usually is small enough to neglect).

In the case of straight pipe, S_{max} exists at the outside of the pipe wall at the point farthest removed from the neutral axis. With curved pipe, S_{max} is taken as the stress in the mid-fibers of the pipe wall at the point of maximum stress shown in Fig. 2 and as determined from *Equation* 9.

Effect of Temperature and Creep

The operation of piping in power plants and oil refineries at temperatures above 750 F has led to the consideration of creep in setting values for safe working stresses due to either fluid pressure or expansion. The line of reasoning applying to the two types of stress is entirely different and may be explained somewhat as follows:

Bursting stress is not relieved by plastic flow; in fact as creep continues the pipe wall will become thinner and the hoop bursting stress will increase rather than diminish. On the other hand, expansion stresses above the creep limit will decrease with plastic flow, since the piping will tend to adjust itself to the form dictated by the hot condition. The result will be that the piping will take a permanent set, thus relieving expansion stresses to a point where further creep does not occur and these stresses remain practically constant from then on. When the piping is cooled down again for any operating reason, it will tend to retain the shape assumed in the hot condition and some bending stress will be set up analagous to that due to cold spring. This is not serious, however, since the stresses in the cold condition can be kept below the initial stresses in the hot condition. On being heated again the line will return to the condition of relieved stress existing after creep had occurred the first time. No harm will result from repeated heating and cooling provided the cycle is not so frequent as to cause failure from fatigue, which possibility is extremely remote in any ordinary pipe line.

Piping for 750 F and Over

The company with which the writer is connected is now basing the design of piping for temperatures above 750 F on the following considerations: (a) transverse bursting stress is to be kept a safe margin within limits which will not produce creep exceeding one per cent in 100,000 hours; (b) the equivalent stress as determined from *Equations* 7 or 10 is not to exceed 15,000 lb. per sq. in. This limit for equivalent stress is considered amply safe for carbon steel pipe having a yield point of not less than 35,000 lb. per sq. in. at room temperature (*A. S. T. M. Specification A*-106) when operating at superheated steam temperatures approximating 850 F. It is believed that the same line of reasoning can be extended to 1,000 F although, presumably, other materials than carbon steel would be required.

While the foregoing analysis represents the consensus of opinion of The Detroit Edison Company's engineers and consultants, it is appreciated that the idea is somewhat novel and perhaps will not be accepted by all. A discussion by those interested in the points involved will be welcomed.

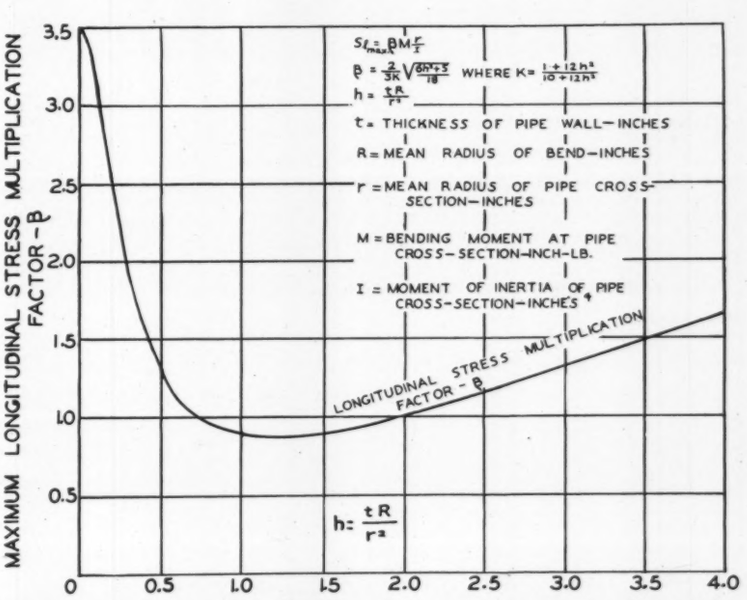


FIG. 6—STRESS MULTIPLICATION FACTOR β FOR DETERMINING MAXIMUM LONGITUDINAL STRESS IN MIDDLE SURFACE OF CURVED PIPE

[10] This simplified solution proposed by Professor Hovgaard has been incorporated in the second (1931) edition of the *Piping Handbook*, from which Fig. 6 is taken.

Air Delivery—An Important Part of Air Conditioning

By Samuel R. Lewis*

NO AIR conditioning system is better than its method of supplying and exhausting air from the conditioned spaces. The realization of the importance of this phase of the air conditioning question leads to some interesting studies in connection with the design of the new research laboratory of the A. O. Smith Corporation at Milwaukee, Wis.

A life-size model of a section of the building was constructed in one of the shops. It was about twenty feet square and eleven feet high. Made of timber lined with heavy plaster board, the model was within a large factory room which was at a normal September temperature of around 75 degrees. The windows of the model were made of light galvanized sheet steel the exact size of the proposed plate glass windows. Outside of the sheet steel were wooden bunkers for receiving crushed ice and salt. See Fig. 2.

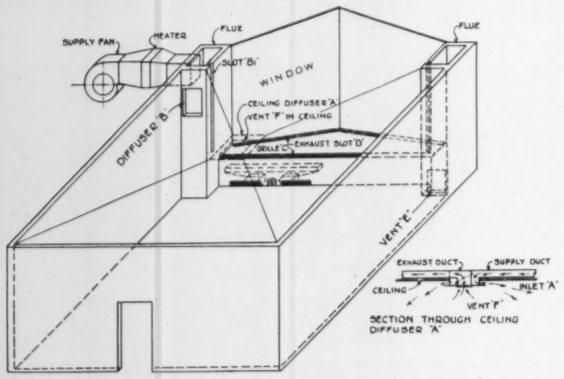


FIG. 1—AN ISOMETRIC DRAWING OF THE TEST ROOM. THE "WINDOWS" ARE OF GALVANIZED SHEET STEEL WITH BUNKERS BEHIND THEM FOR ICE AND SALT. THERE IS A CONVECTION HEATER IN THE WINDOW STOOL, WITH TWO ELECTRIC RECIRCULATING FANS UNDER IT

Both flues may be used as supply or as exhaust or either flue may be used for either function.

Air at any desired temperature may be introduced through the ceiling diffuser *A* or through the diffusers at *B* or through the grille at the window stool *C*.

Air may be exhausted through grilles *D* at the bottom of the glass or through grilles *E* at the flue-bases, or through grille *F* in the bottom of the ceiling diffuser.

The fan and heater could be connected to any of the supply openings, *A*, *B*, or *C*.

Methods of Supplying Air

A supply fan capable of delivering the amount of air allotted to this particular area, with an indirect steam heater for it, was provided.

The fan was arranged so that the air could be delivered easily to:

(A) the ceiling diffuser *A;*

(B) two supply flues, one at each end of the outside wall;

(C) an up-looking duct under an enclosed radiator at the window stool.

The diffuser *A* was in the center of the bay, having deflectors so as to deliver the air horizontally at rather high velocity, through 360 degrees or through any desired part thereof.

Exhausting Air

The room could exhaust its air through:

(D) slots in the window stool close to the glass;

(E) the flues at the ends of the outside wall at the floor line;

(F) an up-looking grille at the ceiling in the center of the diffuser described in (A) above.

The Tests

The following is a resumé of the results of the various tests which were made:

Scheme 1: One inlet as per *B*, 8 ft. 6 in. above the floor, with outlet at *D*, in the window stool. The room was pre-chilled and the windows were thoroughly frosted. The entering air was much warmer than the room air, simulating a winter condition. There was no radiator. Heating was done by the entering air stream. It will be noted that the entering air at about 400 linear feet per minute blows *away* from the outside wall.

Fig. 3 illustrates the unfortunate conditions which resulted. The window-stool exhaust openings were thoroughly inadequate. The entering air spiraled around the room and, crossing the great cold area of the window, pulled along with it great volumes of chilled

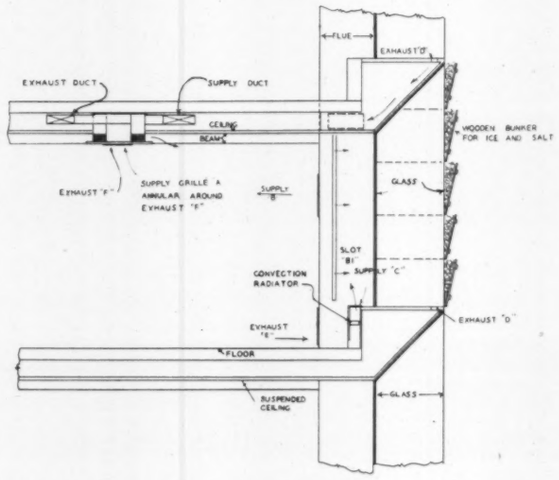
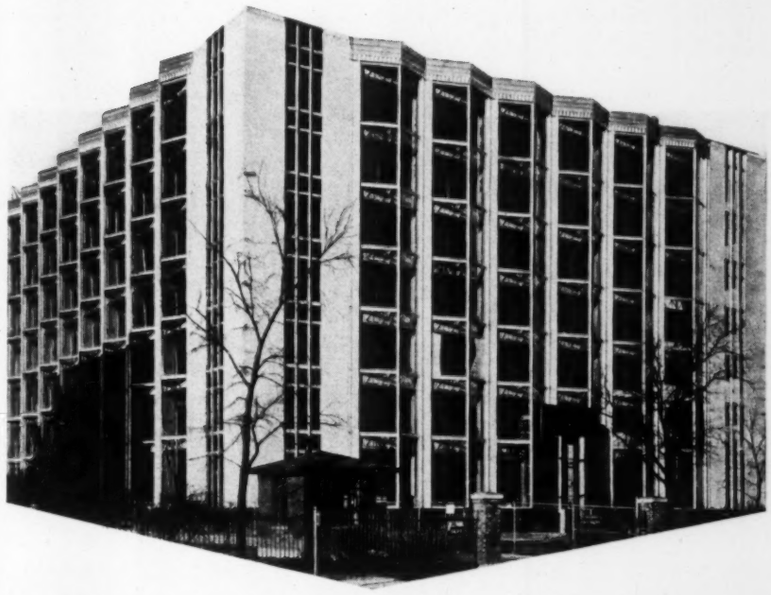


FIG. 2—SECTION THROUGH THE ROOM SHOWING THE ICE BUNKERS OUTSIDE OF THE "GLASS." *A*, *B*, AND *C* ARE POSSIBLE SUPPLY OPENINGS, AND *D*, *E*, AND *F* ARE POSSIBLE EXHAUST OPENINGS. THE CONVECTION TYPE HEATER SHOWN HAS AN ELECTRIC FAN UNDER IT

* Consulting engineer, Chicago, Ill.



TESTS OF AIR DELIVERY AND EXHAUST WERE MADE BEFORE THIS BUILDING WAS CONSTRUCTED

air which made exceedingly uncomfortable conditions in a large zone near the supply flue.

When cooling the room using this arrangement of inlets and outlets, there was less discomfort around the supply flue base than when heating the room, but the cool entering air fell to the occupied zone in the inner parts of the room and caused noticeable drafts.

It would not suffice to use a different scheme of air introduction in summer than in winter.

While this scheme might serve in one large office without cross-partitions it would be awkward if, for instance, there were two private offices in a bay like the model, since in that event one of the private offices would have a supply flue and no exhaust flue, while the other would have an exhaust flue and no supply flue.

Scheme 2 was then tried: The air entered through grille *C*, Fig. 1, at the window stool and was exhausted at *E*. There was no radiator in the room.

In cold weather, with a deflector to shunt the entering air in a lively current toward the glass, the results as to distribution and comfort were reasonably good so long as the entering air was warmer than the air in the room.

When cooling the room, however, there were drafts along the window stool and in the neighborhood of the exhaust opening at *E*. Movement of the air in the parts of the room remote from the window was defective.

Scheme 3: Inlet as per *A*, blowing horizontally through 270 degrees, with exhaust at *F* in the center of the ceiling diffuser.

There was still no heat supplied at the window stools. The air distribution was much improved, but there was still an objectionable draft down the glass, falling over to the floor when heating. When cooling the air, distribution was excellent, though it was somewhat cooler in the interior areas of the room than it was along the exterior wall, which at this time being warm, was of course warming the air which touched it.

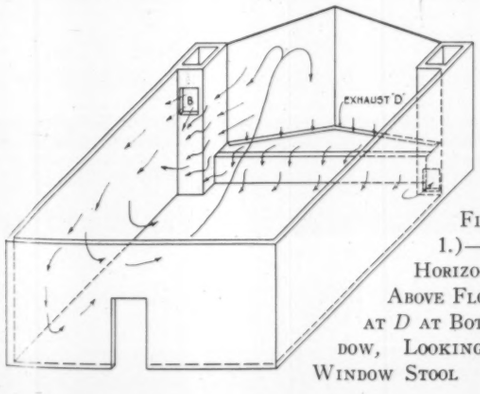


FIG. 3 (SCHEME 1.)—INLET AT *B*, HORIZONTALLY, 8′ 6″ ABOVE FLOOR. EXHAUST AT *D* AT BOTTOM OF WINDOW, LOOKING DOWN IN WINDOW STOOL

Improvement (when cooling), followed the opening of the formerly closed 90 degrees of the diffuser which opened toward the glass.

It seemed clear that the air supply should be delivered toward the heat-gaining and heat-losing surfaces, but it was demonstrated that a warm stream of air alone would not prevent drafts along the window stool.

It was not desirable to install a plant which would require changes in the diffusers whenever cooling was substituted for heating.

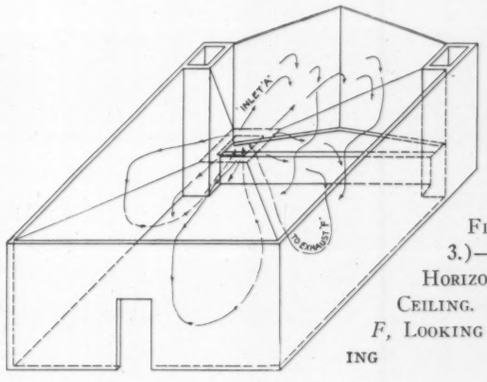


FIG. 4 (SCHEME 3.)—INLET AT *A*, HORIZONTALLY, AT CEILING. EXHAUST AT *F*, LOOKING UP AT CEILING

Scheme 4: The next plan was to add to *Scheme* 3 an enclosed direct radiator at the window stool. When heating, with very frosty windows, this convection radiator caused a sufficiently active upward air current to deflect the down flow of cooled air due to the glass, out into the room far enough that there seemed to be no "discomfort zone."

When as small a part as one foot of the length of the radiator outlet grille was covered up, however, a miniature Niagara of cool air straightway began to fall off the window stool to the floor at that spot.

If the room became warm and if the steam supply to the radiator were shut off, however, the cold draft persisted.

With *Scheme* 4 partitions could be placed at panel points or at intermediate points and every room or half room or any large open space would receive its air supply and its heat independent of partitions, a distinct advantage.

The entering air, using *Scheme* 4, was to be a trifle cooler than the air already in the room the year round. The heaters at the window stools would attend to the warming of the air to room temperature in winter, and the sun would attend to this in summer, with the people and the electric lights always assisting, of course.

The difficulty with *Scheme* 4 was that during winter after the room became warm it would be necessary to shut off or reduce the steam to the radiator; during this period the cool air would come tumbling down the glass, to the discomfort of the occupants.

Scheme 5: This is *Scheme* 4, plus a local electric recirculating fan under each enclosed radiator. This fan drew part of its suction from the former exhaust slots

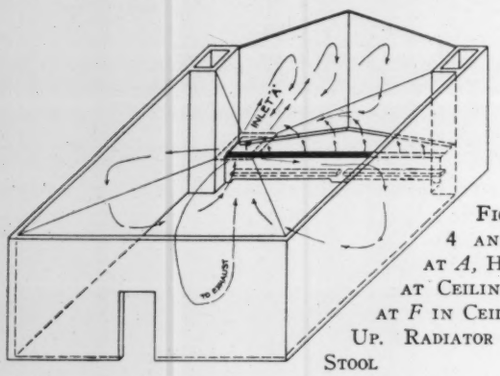


FIG. 5 (SCHEMES 4 AND 5.)—INLET AT *A*, HORIZONTALLY, AT CEILING. EXHAUST AT *F* IN CEILING LOOKING UP. RADIATOR AT WINDOW STOOL

at the window bases and part of its suction from the floor directly under the window stool. The fan was of the double overhung type, practically noiseless, such as is used in unit ventilating machines.

When heating with *Scheme* 5, the effect of the convection heaters at the window stools was increased and a jet of air at rather high speed was deflected toward the glass at an angle of perhaps 30 degrees. There was no objectionable draft at the window stool, when heating or when cooling. Apparently *Scheme* 5 embodied the advantages of the other schemes without their defects.

Scheme 6: This is the same as *Scheme* 5 except that the air supply opening of *Scheme* 1 was used. The same spiral movement of air within the room went into action and cold drafts, winter and summer, were manifest around the base of the supply flue, despite the efforts of the recirculating fans and radiators.

Scheme 7: Since apparently what the room wanted was that the air supply should be delivered toward the glass, *Scheme* 6 was modified to the extent that the air supply was delivered from the flues at each end of the glass area through slots extending from the window stools clear to the ceiling, and about 2 in. wide, looking diagonally toward the glass. These caused two distinct diffusion-currents and resulted in effects comparable in efficiency with those obtained from *Scheme* 5. With *Scheme* 7, the exhaust opening remained in the ceiling in the center of each bay, winter and summer. Thus each half bay, if shut off from its mate, would have an inlet and an outlet and a radiator.

Scheme 8: The radiator was filled with water at an initial temperature of 58 degrees, which ran through the *thermotrap* and back to the plant after having been controlled so as to pick up enough heat from the room to increase the water temperature six degrees.

This water came from a deep well, although it might as well have been supplied by a refrigerating plant. With the fan under the radiator to cause rapid air circulation it became an effective unit cooling apparatus enough

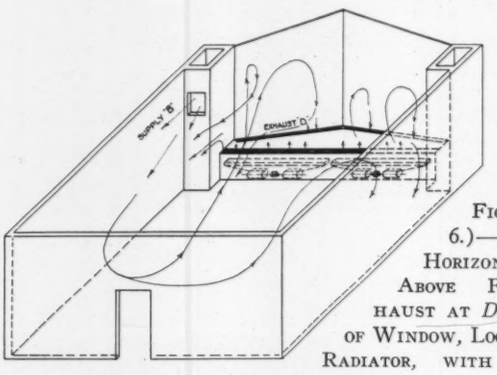


FIG. 6 (SCHEME 6.)—INLET AT *B*, HORIZONTALLY, 8′ 6″ ABOVE FLOOR. EXHAUST AT *D*, AT BOTTOM OF WINDOW, LOOKING DOWN. RADIATOR, WITH FANS, AT WINDOW STOOL

above the normal dew point temperature so that no local condensation of the recirculated air of the room would occur.

(It is realized, of course, that if the air introduced into the room from an outside supply is *too* dry and cool, and if there is too much of it, we will cause a local fog and a local shower of rain, but these limits need not be approached in a plant such as we are studying.)

There is no particular reason, of course, why with non-ferrous radiators and proper gutters we should object to condensation when cooling locally, as described. The diaphragm valves of the normal automatic temperature regulating system could be fitted with a simple built-in mechanism like that for a conventional dual system so that in winter they open to admit heat when the room is too cool, and so that in summer they open to admit more cool water when the room is too warm, thus increasing the direct cooling effect. It probably would be necessary to install manual valves in series with these dia-

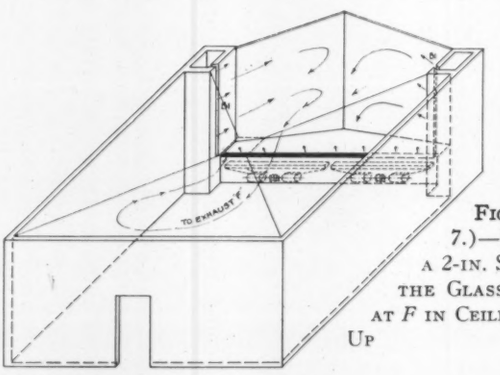


FIG. 7 (SCHEME 7.)—INLET AT *B1*, A 2-IN. SLOT TOWARD THE GLASS. EXHAUST AT *F* IN CEILING, LOOKING UP

phragm valves for equalizing the cool water flow in summer, but these, once adjusted, could be of a type which could not thereafter, without use of tools, be opened more than just wide enough.

The actual experiments ended at this point because of a time limit. The plant was built as per *Scheme* 5, with provision for the installation of the local recirculating fans and for the use of cold water to gain temperature control and efficient cooling.

It is expected a report will be made later concerning the actual construction and operation of the plant.

A General View of the Main Building, Showing Several of the Welding Stations

How Oxygen and Acetylene are Piped in a Manufacturing Plant

By W. A. Slack*

THE oxy-acetylene torch is used at so many points in the main building of the Naylor Pipe Co. in the Chicago district that a comprehensive pipe layout, both for oxygen and acetylene, was required. This plant was built to manufacture spiral steel pipe.

To avoid interference with the usual operations taking place in the manufacture of this pipe, all of the oxygen and acetylene piping was hung from the roof trusses overhead and out of the way of machines and workmen. At various points where torches are required drop pipe lines have been installed, terminating in suitable regulators and hose connections. In all, there are 39 stations, so that 39 welding or cutting torches may be used simultaneously. The pipe line installation is all welded.

The method of installing the piping, the general layout, etc., should be of interest and will, therefore, be described in detail.

Gas House

The gas house is a separate building adjoining the rear of the main factory building. It is constructed of brick and divided into three rooms—one for the acetylene generators and one for the oxygen cylinders and manifold, with a room between for the storage of carbide. The rooms are separated by brick walls and the entire structure is of concrete, brick and steel.

Two 200-lb. acetylene generators have been installed, but the acetylene generator room, the charging platform and the pipe manifold behind the generators was so arranged that a third 200-lb. machine could be installed at any time. Each machine has a total capacity of 1000 cu. ft. of acetylene, which can be delivered into the pipe lines at the rate of 200 cu. ft. per hr., or both machines have a combined capacity of 400 cu. ft. per hr. The arrangement is such that either machine may be used individually or both machines can be run simultaneously.

Acetylene Pipe Line

The header connecting the two machines in the generator house extends along the wall and is of 2½-in. wrought iron pipe. Each generator outlet is connected to this header with a 1-in. line welded to the header. In this 1-in. line a valve is provided at the top of the generator, a union in the line and a valve between the header, so that a generator may at any time be disconnected and removed from the generator house without

* Torchweld Equipment Co., Chicago, Ill.

OXYGEN CYLINDERS ARE CONNECTED TO THE MANIFOLD WITH FLEXIBLE COPPER COILS

interference with the operation of the other machines.

From the header a 2½-in. line extends through the wall to a large 2-in. yoke connecting to two hydraulic check valves. These valves, commonly known as flash backs, are attached to the wall at a level where they can be easily filled with water from time to time as necessity may require. They were so arranged because when such valves are placed up in the roof trusses where a ladder is necessary to reach them, the chances of carelessness on the part of the workmen in keeping them filled with water are greatly increased.

In the yoke portion connecting to the inlets of the flash backs two globe valves are set. In the service lines from the outlets of the flash backs, unions are placed in such manner that the valves may be closed and either flash back removed without interfering with the operation of the other. Two-inch lines extend from the outlets of these flash backs upward to an elevation slightly under the roof trusses and thence along the rear wall of the building to the main service lines under the trusses. One of these lines extends lengthwise of the building about 30 in. from the center line and the other lengthwise of the building along one side wall. Both lines are suspended from the roof trusses with hanger brackets.

For a distance of 100 ft. both lines are 2-in., reducing to 1½-in. for a distance of approximately 80 ft. From these main lines horizontal laterals extend perpendicular to the center line of the building. Sixteen of these laterals are equipped with two drop pipes, each extending downward to an elevation about seven feet above the floor.

The laterals for approximately 20 ft. each are of 1-in. pipe, reducing to ¾-in. and terminating in ¾-in. globe valves and ¾-in. by ¼-in. by ¼-in. malleable tees to which the acetylene regulators are attached.

The acetylene pipe lines were inclined slightly toward the hydraulic water seals in the rear of the building in order that all condensation should drain back to these valves rather than pocket in the pipe. All angles in the pipe were made of bent pipe of long radius. All joints were welded and all pipe was wrought iron standard black pipe.

Disposal of Sludge

In the floor of the generator room a trough was constructed directly under the sludge outlets of the generators, with a piece of pipe attached to each outlet in such manner that sludge will flow into the trough without splashing over the floor. The troughs lead to two sludge pits outside of the building. These pits are 8 ft. by 6 ft. by 6 ft., each equipped with a tilting stand pipe connected to the outlet to the sewer.

The stand pipe is a length of pipe attached to a loose threaded elbow which can be raised or lowered with a chain to any elevation within the pit. The sludge settles to the bottom of the pit and the pipe is used to drain off the clear water above. As the elevation of the top of the pipe can be varied, all of the clear water can be drained away in this manner from time to time. Usually about once a month the sludge will have to be removed by hand and disposed of in as profitable a manner as possible.

Sludge can be used for many purposes and has a market value if available in sufficiently large quantity.

The pits arranged with the stand pipe outlets permit only clear water to flow into the sewer.

Carbide Storage

The middle room of the gas house is arranged for the storage of carbide. The room is 13 ft. in length by 9 ft. 6 in. in width and will contain an ample supply of carbide.

In accordance with the underwriters' requirements, this room is separated from both the oxygen manifold room and the acetylene generator room with brick walls. All doors and windows of the gas house open to the outside and none of the rooms are connected. Electric switches for all of these rooms are placed on the outside of the building.

Oxygen Manifold Room and Oxygen Pipe System

The oxygen manifold room is 13 ft. in length and 9 ft. 6 in. in width. A special manifold extends around three sides of the room, securely fastened to the walls with brackets at a suitable height for the convenient connection of the oxygen cylinders. The manifold is arranged for 12 cylinders on each side and is constructed of 1½-in. double extra strong wrought iron pipe.

Fittings are set into this pipe at suitable intervals and connected to the oxygen cylinders with flexible copper coils. Both pipes terminate in a valve manifold to which two regulators are attached. The outlets of the two regulators connect to a 2-in. extra strong pipe, extend upward and through the wall and terminate in a 1½-in. header at the end of the shop, which follows and parallels the acetylene line.

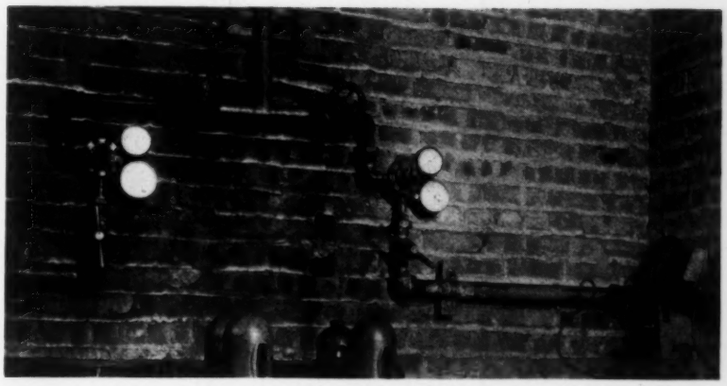
The oxygen manifold is so arranged that either bank of 12 cylinders may be discharged into the service lines using either one or both regulators.

The valves are so arranged that either regulator may be removed from the line while the other regulator is still in operation. By closing the center valve between the two regulators, an empty bank of cylinders may be drawn off down to the line pressure, while the other bank of cylinders is providing the main supply of oxy-

gen. The arrangement of valves in the oxygen manifold is comprehensive and enables the manifold to be utilized in a very convenient and efficient manner.

From the header across the end of the building the oxygen lines extend for a distance of 100 ft. paralleling the acetylene lines and being 1½ in. in diameter. From this point to the end of each line the diameter is 1 in. All branch lines to the drops are ¾ in. diam., terminating in ½-in. drops to the regulators.

All oxygen pipe is extra strong wrought iron pipe with all joints welded. All angles are made of pipe bent on a long radius to avoid as much as possible resistance to the flow of gas caused by right-angle turns.



THE OXYGEN VALVE MANIFOLD

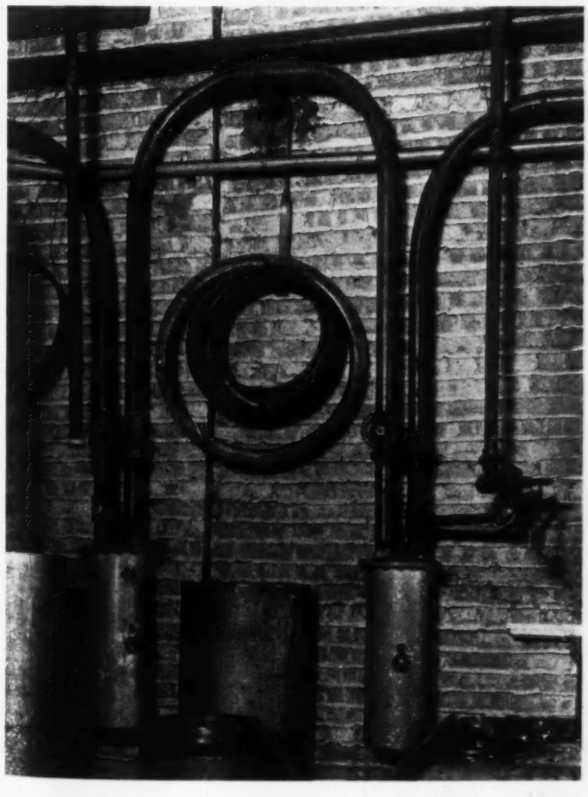
There are 39 oxygen stations located close to the acetylene stations and each is equipped with an oxygen line regulator. The drops terminate in ½-in. heavy duty globe valves and ½-in. by ¼-in. by ¼-in. malleable tees to which the regulators are attached.

At the end of the building in the header between the two main oxygen lines are several fittings which have important functions. Three feet each side of the main oxygen pipe line, extending from the oxygen manifold room through the wall of the building, are located heavy duty globe valves with which either line may be entirely shut off. Three feet beyond these valves a check valve is inserted in each branch line, and in like manner three feet beyond these check valves is a relief valve set to blow off at 150 lb. pressure. The function of these check valves and blow-off valves is to prevent any possibility of a reversal of the flow of gas toward the manifold. The joints between the pipe and these fittings are threaded joints and were set up with litharge and glycerine and allowed to set for 48 hours before being tested.

All oxygen pipe was heated in a bonfire to a red heat in order to burn out any oil or grease which might be located within the pipe. The pipe was then washed with a hot, extra strong solution of caustic soda. This treatment of the pipe assured that its internal surface was absolutely clean and entirely free from oil, grease or other foreign matter. In erecting oxygen pipe lines this is of particular importance and should be given the careful attention of every engineer in charge of such work. Oxygen should never be turned into an oily or greasy pipe line, and care in this respect is essential to safety. As is well known the country over today, oxygen and oil or grease form an explosive compound which must be avoided.

The entire pipe line system, including both the oxygen and the acetylene lines, was tested to a pressure of 125 lb. before being put in service. All joints were gone over with soap suds to find any possible points of leakage and these were repaired with the torch.

FLASH BACKS ARE ATTACHED TO THE WALL AT A LEVEL WHERE THEY CAN BE KEPT FILLED WITH WATER EASILY



Tight System Essential

When piping acetylene, the line must be absolutely tight because leaking inflammable gas should never be permitted in a factory building. An oxygen line should be tight because oxygen is too expensive a gas to waste in this manner.

Low Vacuum on Heating Return Lines

Quite often when the vacuum is low on an extensive system of heating return lines, the excuse is, "The vacuum pump is no good," when it is only necessary to get busy and locate leaks in the return pipe lines or steam traps, obstructions in return pipe connections, drip valves open to the atmosphere, and similar faults.

A few traps blowing through will fill the return lines with steam and the heat from this steam will cause other thermostatic traps to close and interfere with the heating at other points. Frequently in a heating system consisting of several buildings, when one building fails to heat properly, the trouble may be in the equipment of an adjacent building. Just because a radiator is hot is not always a true sign that it is heating efficiently. It may have a steam trap leaking that will cause trouble to other units; it is also a wasteful and expensive practice.

Air Conditioning for Homes —Cooling

By S. M. Anderson*

THE cooling of a home during the summer months offers other problems to be dealt with aside from those pertaining solely to humidity control. The same apparatus used for cooling purposes may be adapted to heating and humidifying, and in fact a system which was designed for summer cooling but which could not be used for winter air conditioning would be most ineffective in cooling. However, the converse is not necessarily true, as there are many homes which have been well laid out for heating and humidifying that can never be properly cooled by their present equipment.

Usually the volumes of air circulated for cooling purposes run to much larger quantities than for heating and humidifying. For example, where a home might require five thousand cubic feet of air per minute for cooling, it would be possible to heat and humidify it with only two or three thousand cubic feet of air per minute. But in general the use of large quantities of air both summer and winter makes it possible to obtain uniform temperatures about the house, making each room less affected by exposure. Introduction of the air in winter time would then be at lower temperatures, so that the winter system becomes a "warm" air rather than a "hot" air system. In other instances it has been found to be more desirable to the home owner to provide a fan apparatus with two speeds so that full cooling volumes can be circulated in summer time and smaller volumes of air in winter at the reduced speed of the circulating fan.

The Amount of Cooling

The amount of cooling required is variable, depending upon individual preferences. There are some who desire a system that will cool all of the rooms of a house; others are satisfied to condition the air in only the dining and living rooms or possibly in only one room. Moreover, there are as many ideas as to how much the temperature of the refrigerated rooms should be reduced below the outside temperature as there are persons wishing to install cooling systems. In such matters, the air conditioning engineer is usually guided by the wishes of the individual.

In some instances an acceptable system can be obtained by providing adequate air circulating machinery which operates only in the coolness of the late evening, blowing the hot air of the daytime out of the house and replacing it with cool air from the outside after the sun has gone down. Such an arrangement cannot, of course, equal the effectiveness of a true refrigerating system.

A person building a new home should decide in advance how much air conditioning he desires, i.e., whether he will want refrigeration in all or a part of the house, or whether he will be content with plenty of air movement without refrigeration. He should also decide before-hand upon the system for supplying heat and humidity for winter operation. This will determine the size of the duct work and fan equipment before the walls of the house have been plastered, and when it is too late for changes to be made. The person who is remodeling a house has a more difficult problem to solve, and one which cannot always be handled without considerable expense in tearing out walls and flooring.

Provisions May Be Made for Future Cooling

In a new home located in one of our southern cities a system was put in strictly for humidifying and heating purposes, but with the idea that at some future date cooling apparatus could be obtained and added to the present equipment to give a complete refrigerating system. Fig. 1 shows the four section, gas-fired heater in the basement of this house, with a humidifying pan directly over the middle two sections of the heater. When the cooling system is installed, this humidifier pan will be abandoned and a complete humidity control installed in conjunction with the combined summer cooling washer and winter humidifier.

Directly behind the heater is the plenum room in which are located the fan equipment and motor, and into which all of the return ducts from the rooms of the house are led. The door, normally shut, opens into this plenum room simply for purposes of observing the operation of the fan and motor. When the cooling apparatus is added to this equipment, the return ducts will be led through the cooler into the plenum chamber, and all of the air will thence be blown through the heating elements into the rooms of the house. The heater will not be in operation during the summer time except for tempering purposes. Fig. 2 shows fan and motor connected to a heater and ready for the erection of a plenum chamber. This is not a picture of this installation but shows identical equipment.

It is evident that care is necessary to insure the quiet operation of the moving

* Research engineer, Air Conditioning Department, B. F. Sturtevant Company, Boston, Mass.

A good night's sleep no matter how hot and humid the outdoor atmosphere may be—a pleasantly cool dining room when the sun is shining its hottest at noon, or in the warm evenings after it's done a full day's work—comfortable living rooms all day—a kitchen that is not unbearable; these are but a few of the advantages an air-conditioned home can offer.

The public is keenly interested in the air conditioning of residences these days. Undoubtedly the manufacture and installation of air conditioning equipment for homes will be an industry of ever-increasing importance during the next few years.

S. M. Anderson, research engineer, has something to say about cooling homes in this article—something you'll find of value.

apparatus, because it is all located directly in the flow of air where the slightest sound would travel through the duct work to the house itself. The fan, in such cases, should be designed to run at the proper rate of speed in order to assure a uniform distribution of air at all times, without disagreeable noise. Sharp noises such as motor hums and mechanical ticking sounds should be eliminated from the apparatus or isolated from the system; and the motor should be a type which does not interfere with radio reception and will not cause a change in the brilliancy of the electric lights of the house when it is turned on or off.

The Spray Cooling Unit

Fig. 3 is a diagram of one type of spray cooling unit for household use. As the air from the house is drawn through this unit it passes through the baffles and filters, at which time it first comes into intimate contact with the spray water. The water from the spray nozzles penetrates by the force of its own pressure through the interstices of the filter to the baffles, and is thrown back again by the air velocity onto the filters as it loses momentum in its original projected path against the air flow. In this way the air is scrubbed and washed by the wet surfaces of the filters, freeing it of dust and dirt, and the filters are continually flushed by the spray water carrying this sediment down into the drain tank. The air then passes directly through more atomized water from the nozzles and is further washed and cooled. A portion of the spray water from the nozzles is fine enough to be carried with the air flow and never reaches the filter at the end of the washer into which the air is introduced. This water is carried against the second set of filters and eliminators, where it is caught and discharged into the drain tank, carrying with it more dust and dirt.

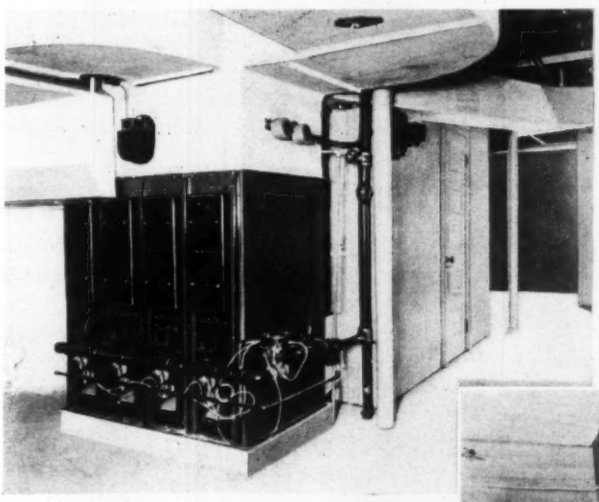


FIG. 1—FOUR SECTION, GAS FIRED HEATER INSTALLED IN THE BASEMENT OF A HOME. PROVISIONS WERE MADE ON THIS JOB FOR THE FUTURE INSTALLATION OF COOLING EQUIPMENT

Incidentally, with an air conditioning system, there will be very little occasion for the use of a vacuum cleaner in the house, as long as the system is kept in operation.

Water Necessary

Chilled water for the spray nozzles must be continuously furnished by additional apparatus having a goodly supply of "potential coldness." This may take the form of a refrigerating system of some simple design, or, in some localities, of a well driven into the ground and tapping a supply of cold water conserved from the rains and snows of the previous winter. Spring water and city reservoir water have possibilities for use in some places, but on account of the large quantities needed, it is generally impracticable to rely upon these sources.

When it is considered that the average house would require approximately seventeen gallons of water per minute for adequate refrigeration (all of which must be wasted if drawn from a city reservoir) it is at once apparent that a large number of homes using water at this rate would seriously endanger a city's supply during the hot summer months when the rainfall is often scant. Furthermore, by the time water from a common reservoir has been stored in auxiliary standpipes and inter-connecting pipe lines for a period of time, it becomes warmed above the temperature where it can be utilized for effective cooling. It is not uncommon for such water to reach a temperature of 60 F to 65 F during July and August, which is about ten or fifteen degrees too high. For these two reasons, the possibilities of the use of city water as a cooling medium are remote, except in connection with refrigeration where the higher temperature water can be passed through a condenser.

Use of a Deep Well

Deep well water seldom goes over 55 F, even at the end of the summer, and in localities as far south as Washington, D. C., this offers a solution to the problem of household cooling, provided one is fortunate enough to strike water at a reasonable depth. A deep well supply of water is usually simpler than refrigerating equipment. The question of noise from the well pump need not prove a troublesome one if care is taken to isolate the pumping apparatus and prevent it from transmitting its vibration to other parts of the house. It is usually better to locate the pump outside of the house, running a short pipe line underground to the cooler within the house itself, but experience has shown that a successful installation can

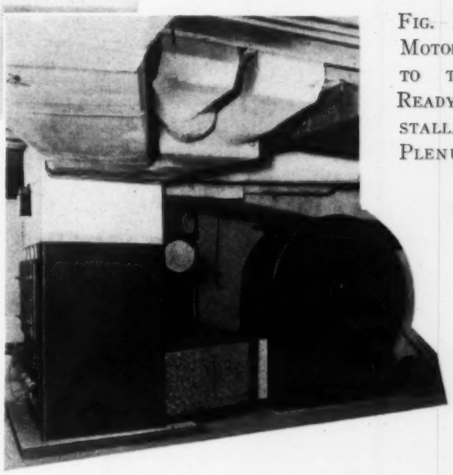


FIG. 2—FAN AND MOTOR CONNECTED TO THE HEATER READY FOR THE INSTALLATION OF THE PLENUM CHAMBER

be made having the pump and well within the basement of the house.

Refrigerating Equipment

It is not the purpose of this article to go into great detail regarding the merits or defects of mechanical refrigeration. There are many good machines on the market—some more quiet than others, some occupying more space than others, some operating at prohibitively high speeds, others at normal motor speeds and yet others at low speeds which require elaborate inter-connecting gears or belting to step-down the revolutions per minute from the rate of a standard alternating current motor. Although no chilled water is wasted in a system of this sort, it becomes necessary to get rid of the heat from the house by some means. This may take the form of a cooling tower or fan and washer system which draws air in from the outside and warms it by contact with the condenser water containing the heat given up by the refrigerant, discharging this air again to the outside at a higher temperature.

Dampers for Regulation Imperative

For perhaps forty days of the summer, depending upon the locality, actual refrigeration would be necessary for comfortable conditions. During the remainder of the warm weather it might well be sufficient to operate the fan system alone, keeping a good circulation of air passing through the house, drawn from the outside through the basement and discharged to the outside from the open windows on the upper floors, with the spray nozzle cooler operating on a recirculating water basis, cooling by natural evaporation. It is, therefore, imperative that the movement of air be regulated by a system of dampers manually operated, so that, at will, either a complete recirculation of the air in the rooms to be refrigerated can be obtained, or else the same volumes of air can be drawn in from the outside and discharged through the windows. Whenever a hot summer day is succeeded by a cool evening, the apparatus can be run on the setting of dampers providing outside air. On the other hand, during periods of heat when refrigeration is necessary, the system may be set to the recirculating air cycle with all of the windows tightly closed.

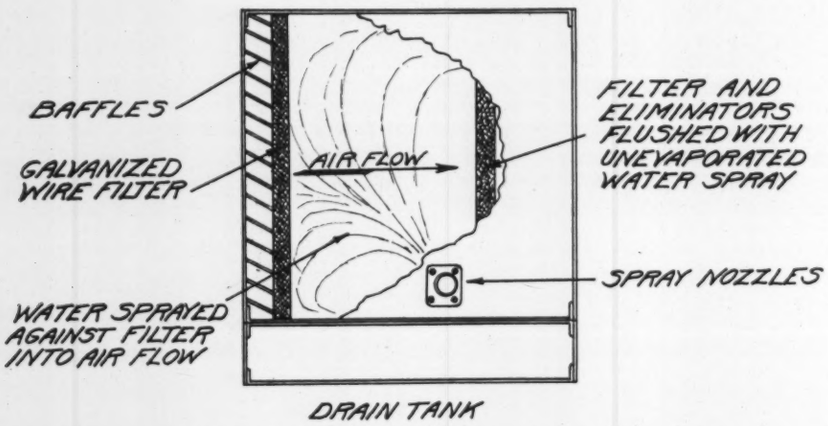


Fig. 3—Diagram of One Type of Air Conditioning Unit Which Provides for Humidifying, Filtering, Cooling and Scrubbing the Air

There are ordinarily so few people in a home that it is not considered necessary to admit the usual amount of outside air into the recirculating system. As a matter of fact, enough natural leakage will usually occur to amount to, say 20%, and the only advantage gained in bringing in the air mechanically would then be that a definite, known quantity of outside air can be admitted. It is estimated that the mechanical admission of outside air puts the house under a slight pressure—enough to balance out and prevent leakage through the walls due to natural windage. Just what effect can be obtained in such manner is a debatable question, and depends largely upon the construction of the particular house.

The question of automatic control in the summer time is usually not as serious as in the winter. There are no chilled surfaces in the rooms of the house where moisture will condense, unless it is on the *outside* of the window panes where it will do no harm. There would, therefore, be no great advantage in attempting to utilize the winter humidity control apparatus.

By maintaining a constant temperature of water in the air conditioner, which is a simple matter with well water

A Completely Air Conditioned Baltimore Residence, Humidified in Winter and Cooled and Dehumidified in Summer. The Equipment Has a Cooling Capacity of Six Tons. Architects, Palmer & Lamdin; Builders, Thos. Hicks & Sons. The Heating Contractors Were Wm. E. Wood Co.

and even less difficult with mechanical refrigeration, a constant dew point within the house is approximated, and with properly designed by-passing dampers or tempering heaters, a uniform living temperature can be assured.

In a well insulated house there do not occur the rapid fluctuations of temperature inside the walls that are observed in the fickle changes of the outside weather. In other words, there is within the house a thermal inertia producing a tendency for the temperature to remain steady from day to day, independent of outside changes. Ordinary dry bulb temperature controls may therefore be used.

Generally speaking, it is desirable to use as simple a system of controls as possible. For example, the few necessary switches may be so arranged that they can be thrown in one direction for summer operation and in the opposite direction when it is desired to operate the winter heating and humidifying plant. Thus, while the air conditioning system as a whole appears complicated and involved, its operation can be made very simple.

Lint Collector for a Laundry

By W. H. Pierce*

Question: The exhaust from a clothes dryer installed in the laundry of a hospital runs through the wall to the outside. The ground surrounding the exhaust is literally white with lint from the dryer.
I would appreciate receiving advice which will aid me in suggesting a remedy for this condition.—H. L. Nunamaker.

A DISAGREEABLE feature connected with the tumbling of clothes in the modern laundry is the lint which is given off during the process.

Plants that are located in industrial or sparsely settled districts need not be as particular about what happens to the lint as plants located in residential sections of the town. Lint in laundries thus situated is sometimes allowed to pass out by means of a duct through the roof. It often gathers in drain spouts, clogging them up and creating considerable difficulty during or following a rain storm.

As a means of eliminating this trouble and keeping the vicinity immediately surrounding the laundry free from lint, the lint collector shown in the sketch may be constructed. This lint collector may be made by constructing a frame-work of angle iron and strip metal 1 in. x $\frac{3}{16}$ in. in size. The frame-work can be riveted together as shown.

The outside of the box, with the exception of the right end, should be made of galvanized metal which will withstand the corrosion of atmospheric conditions. The right end, which is shown hinged, should be made of a frame-work of 1 in. x $\frac{3}{16}$ in. strip galvanized metal forming a frame supporting a $\frac{1}{4}$-in. mesh screen. When in operation, the right end will be placed up against the end of the metal box and held in place by dropping the cover, which is also hinged, into a closed position.

There are four baffle plates supported at each end by grooves made of two 1-in. x $\frac{3}{16}$-in. angles. This type of construction is used so that these baffles may be raised upward and removed from the lint collector during the cleaning period.

The lint will be carried into the lint collector by the exhaust fan on the tumbler. After entering the box, the lint will tend to travel in a straight line. It will strike the first baffle, which will be made of a frame-work consisting of 1-in. x $\frac{3}{16}$-in. strip metal and $\frac{1}{2}$-in. mesh screen. The lint that is not caught on the first baffle will be caught on succeeding baffles.

Note that the baffles are made of $\frac{1}{2}$-in. screen, while the right end of the box is made of $\frac{1}{4}$-in. screen, so that the $\frac{1}{2}$-in. mesh will collect the coarser lint, whereas the $\frac{1}{4}$-in. will act as the final screen and catch anything not removed by the former baffles.

Leave Space Around Baffles

Space should be left at either the top or the bottom of the baffles, depending upon their location, which will allow the air from the fan to pass on through the box without creating any great amount of back pressure on the fan. If the baffles were to extend the full depth of the box, they might become completely closed off, thereby eliminating the passage of air out of the lint collector. The front side of the lint collector is shown cut away in part to give a better view of the baffle construction.

To Clean the Collector

To clean the lint collector, raise the top, remove the four baffles by pulling upward, drop the $\frac{1}{4}$-in. mesh at the end, and it will be a simple matter to rake out the lint deposit in the bottom of the collector. The lint may be removed from the baffles by a dull-toothed rake or heavy wire brush.

The period for cleaning this device will depend largely upon the size it is made. If its dimensions are 3 ft. high by 3 ft. wide and about 4 ft. long, it should not be necessary to clean it oftener than every other day.

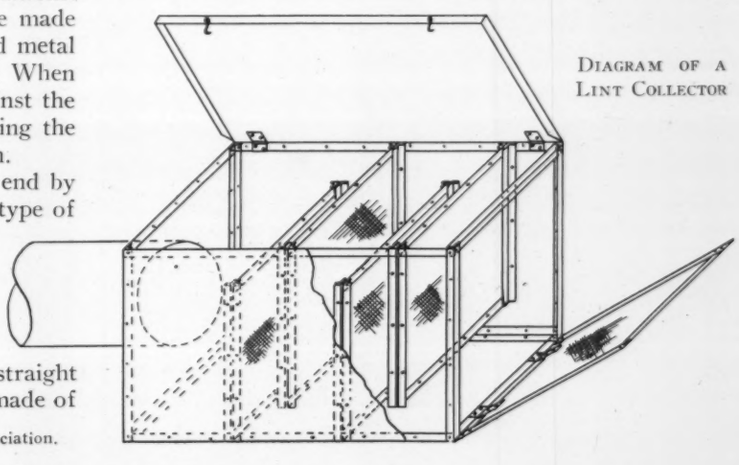


DIAGRAM OF A LINT COLLECTOR

* Department of Engineering, Laundryowners' National Association.

Supports and Anchors for Piping

—Three Typical Problems

By G. W. Hauck

THE proper supporting and anchoring of connections to turbines and steam driven auxiliaries is too often ignored or at least given very little consideration in the designing and installation of piping. The manufacturers of turbines and auxiliaries are of late years emphasizing the fact that piping must be installed so as not to impose piping strains upon the throttle valves, and correctly so, as usually the piping can easily be installed to prevent undue strains or loading on the throttle valve or the machine itself. To illustrate and discuss this point a number of sketches of typical and practical installations have been prepared and will be considered in this article.

Anchoring Turbine Header

Figs. 1, 2 and 3 show the plan, elevation and end elevation, respectively, of the header piping and connections to two small turbines. In this job, there is a 6-in. steam header with 5-in. and 3-in. connections to two turbines for the high pressure piping. From the boiler header to the turbine piping header shown for the two machines, there is about forty lineal feet of piping with a right angle turn and, to keep the expansion strains from being imposed upon the branch connections and through them to the turbines, it will be noted that the 6-in. turbine header is anchored adjacent to the 6-in. steam separator and before the first branch is taken off the header. This keeps piping strains from the branch connections and forces the strain away and into a bend and long tangents in that piping from the boiler room and which is not shown on the sketch. It will be noted that the turbine piping header is mounted on two steel brackets with an anchor on one bracket and a roll support on the other bracket.

The 3-in. and 5-in. steam branches to the turbines are anchored at the floor by means of base elbows, which have the bases anchored in concrete piers, which are a part of the turbine foundation. Flexibility between the header and base elbows is obtained with the quarter bends shown and which have good tangents. In connections of this sort it is necessary that the piping be accurately aligned to be certain that the face of the flange on the base elbow connecting to the flange on the throttle valve is absolutely in line without strain and the base flange then grouted for the anchor.

Exhaust Connections

In the exhaust connections, the copper expansion joints provide sufficient flexibility, as the header is carefully supported in the basement from the floor above, as shown in the elevation, Fig. 2. These hangers on the exhaust header are equipped with rods and turn-buckles for adjustment. Just beyond the piping shown in the drawing an anchor has been provided by welding angle iron to the piping and connecting to a convenient I-beam.

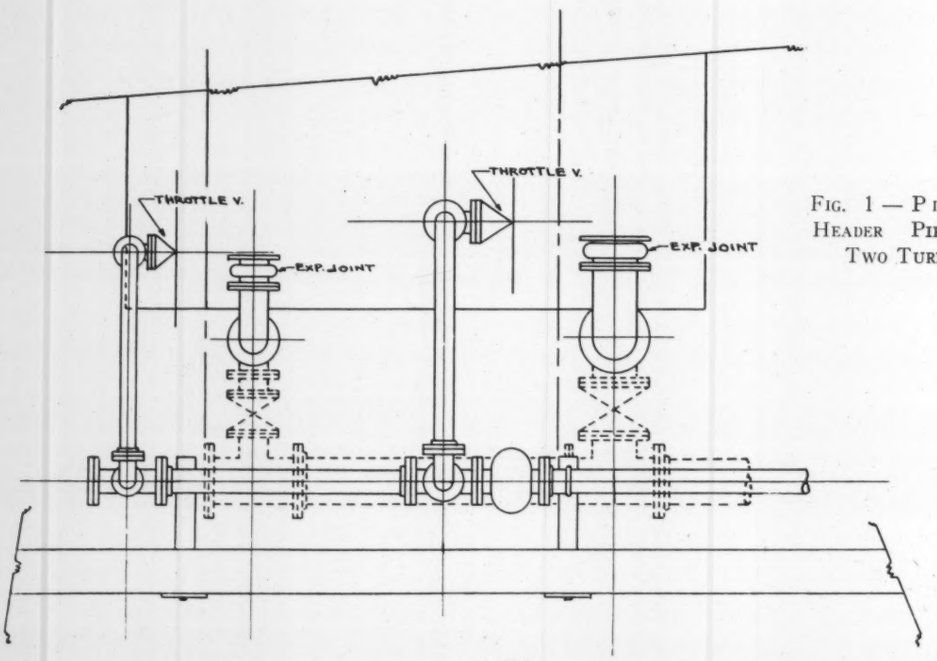


FIG. 1 — PLAN OF HEADER PIPING TO TWO TURBINES

for operation from the turbine room floor. The chain wheels on the high pressure header valves with the floor stand operation of the exhaust valves provide

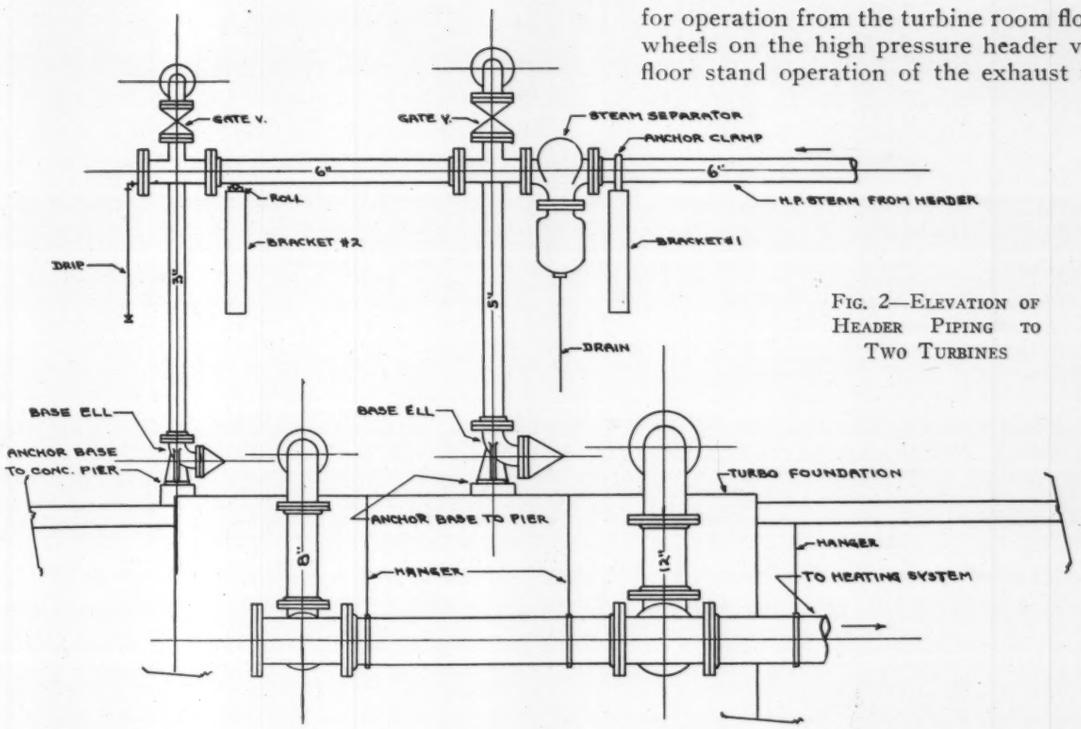


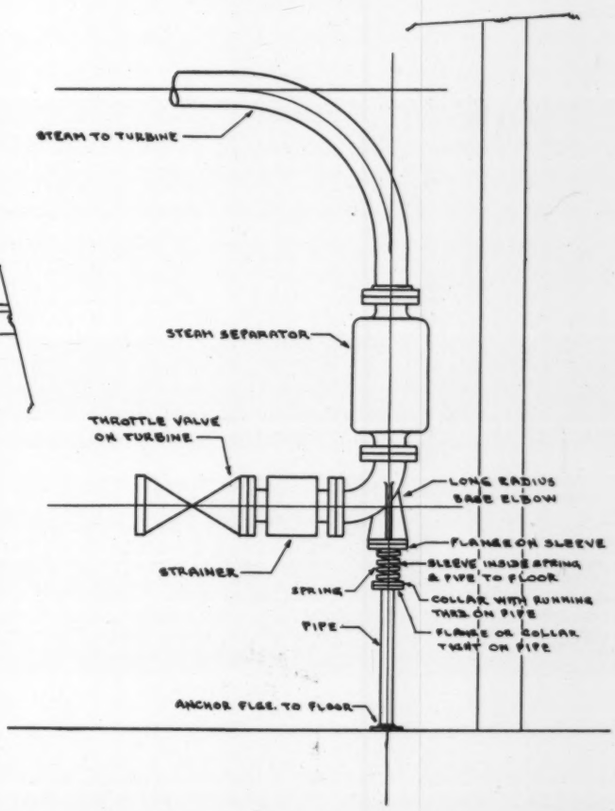
FIG. 2—ELEVATION OF HEADER PIPING TO TWO TURBINES

complete control for the operators of all valves, including the turbine throttles, from a central location.

Connection to Turbine from Overhead Main

Fig. 4 shows an 8-in. piping connection to a turbine, which had to be made from an overhead steam

FIG. 4—EIGHT-INCH PIPING CONNECTION TO A TURBINE FROM AN OVERHEAD MAIN



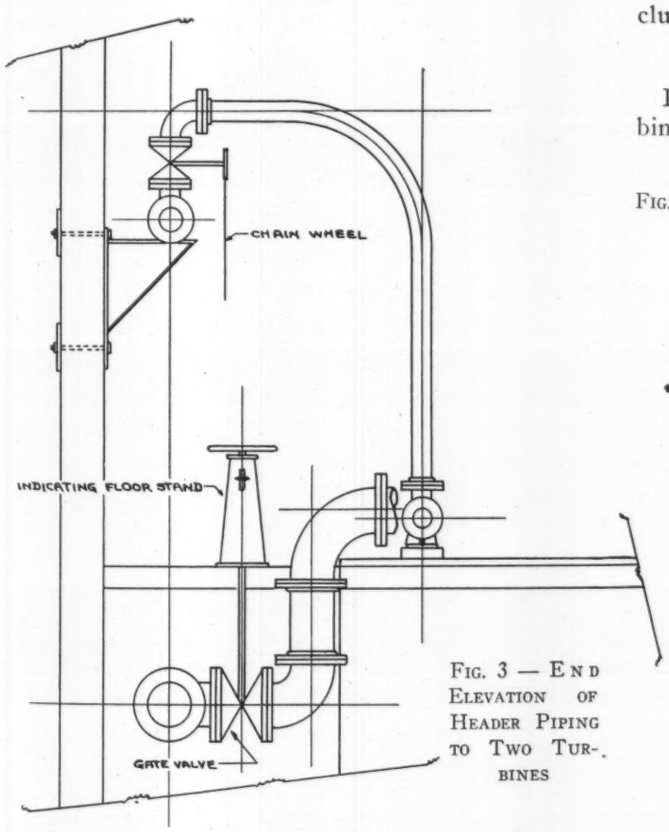


FIG. 3 — END ELEVATION OF HEADER PIPING TO TWO TURBINES

Valves Controlled from Central Location

An interesting side-light is developed in this installation, in that chain wheels have been provided on the gate valves in each connection from the header to turbines. The exhaust valves under the floor in the basement are provided with indicator floor stands

main. In this particular job, it was necessary to provide a spring support to carry the weight of the turbine throttle and strainer, and the weight of the adjacent piping and separator. To accomplish this, a long radius base elbow was used and a spring support provided as shown. Inside the spring there was installed a sleeve made of tubing, machined to slip inside the pipe of the support. This tubing sleeve is threaded tightly into the flange at the top and bolting to the base flange on the elbow. The pipe in the support has a long running thread with two collars, the upper collar being adjustable on the running thread to provide more or less tension on the spring for final adjustment, and the lower collar being a locking collar. This is a spring support which can be easily made and is quite effective.

Fig. 5 illustrates another method of providing support to prevent strains on throttle valves. In this particular job, the support was made by using stock material on hand and, in the absence of a base elbow, a tee was used. A blind flange, faced on both sides, was installed below the lower flange of the tee and to this was bolted a reducing flange with a pipe support to the floor. Base elbows are not always made or carried in stock smaller than 4-in. and so this method of using a tee with blind filler flange can often be used for connections smaller than 4-in., where a quick installation is required. This same construction using a tee can be equipped with a spring support where needed, similar to the layout shown in Fig. 4 for a base elbow.

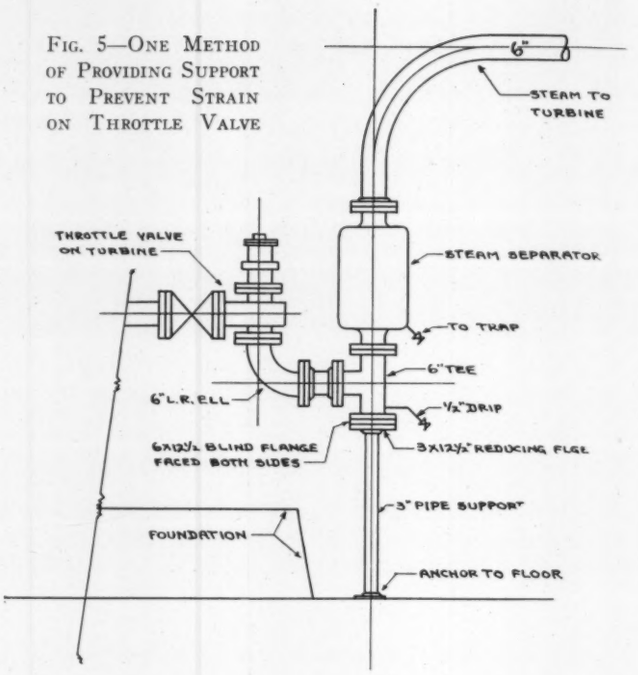


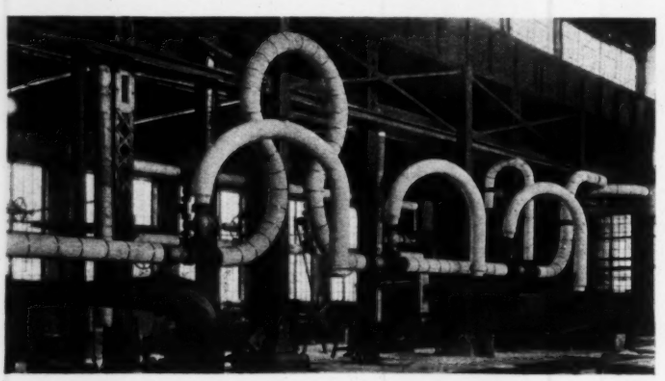
FIG. 5—ONE METHOD OF PROVIDING SUPPORT TO PREVENT STRAIN ON THROTTLE VALVE

Bulletin on Flow of Brine Issued

Bulletin No. 222, "Flow of Liquids in Pipes of Circular and Annular Cross-Sections," by Alonzo P. Kratz, Horace J. Macintire, and Richard E. Gould, has been issued by the Engineering Experiment Station of the University of Illinois. It constitutes a report on the continuation of the work the results of which were published in Bulletin No. 182.

The previous bulletin dealt with the determination of the relation between the friction factor and Reynolds' number when commercial calcium chloride brine is circulated in standard wrought-iron pipes under the conditions encountered in refrigeration practice. The present bulletin deals with the determination of the relation between the friction factor and Reynolds' number for two sizes of standard wrought-iron pipe, with two fluids, water and calcium chloride brine, the relation between the friction factor and Reynolds' number for channels of annular cross-section, with water and commercial calcium chloride brine, and the head loss resulting from the use of standard cast-iron elbows in pipe lines conveying commercial chloride brine.

As a result of the investigation it was found that the use of the Reynolds' number afforded a means of correlating friction factors over a wide range of conditions and was independent of the fluid used in cases in which channels with geometrically similar cross-sections were involved. In the case of the two commercial pipes used in the investigation the difference in the roughness of the rubbing surfaces did not introduce deviations in the friction factor of sufficient magnitude to be taken into serious consideration. The use of the Reynolds' number for correlating friction factors in cases of channels in which the cross-sections were not geometrically similar was found to be not strictly justifiable, although for any particular section it served to correlate the friction factors with the temperature, velocity, and viscosity for all fluids. The Reynolds' number was not applicable for correlating the head lost in elbows; the loss in head in an elbow was largely influenced by the presence of eddies and cross currents rather than by surface friction.



Copies of Bulletin No. 222 may be obtained without charge by addressing the Engineering Experiment Station, Urbana, Illinois.

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

MAY 1931

Material in this Section prepared by Publication Committee: W. A. Rowe, *Chairman;* E. O. Eastwood, and F. D. Mensing : A. V. Hutchinson, Manager of Publications

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Study of Performance Characteristics of Oil Burners and Low-Pressure Heating Boilers

By L. E. Seeley[1] and E. J. Tavanlar[2], New Haven, Conn.
MEMBERS

This paper is the result of research conducted at Yale University, New Haven, Conn., in cooperation with the A. S. H. V. E. Research Laboratory and the American Oil Burner Association.

THE object of this study was to determine the performance characteristics of oil burners and low-pressure heating boilers when operating under various conditions. This report includes the results of numerous tests made with several oil burners and low-pressure heating boilers.

Apparatus Used in the Study

The apparatus tested to date comprises four oil burners and four low-pressure heating boilers which are designated by letters *OB* and *B* respectively. For example, *OB 1—B 2* means the combination of oil burner No. 1 and boiler No. 2. Fig. 1 is a photograph of the set-up. A brief description of the boilers tested is given in Table 1.

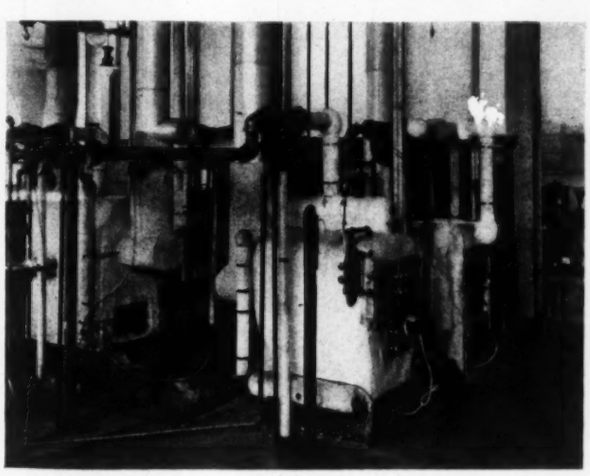


FIG. 1—GENERAL VIEW OF SET-UP OF APPARATUS

Test Set-Up and Procedure

The test set-up and procedure was in substantial agreement with the A. S. H. V. E. Proposed Code for Testing Steam Heating Boilers Burning Oil, edition of March, 1931. See JOURNAL of the Society (Heating, Piping and Air Conditioning), April, 1931.

Fig. 2 shows a diagram of the boilers. In addition to the usual equipment it should be mentioned that each boiler was equipped with an automatic draft regulator. An audiometer was used for sound measurements. The instrument is calibrated in terms of sensation-units, a standardized unit of sound. By means of a special ear piece the sound from the burner and from the audiometer enters the ear and the sound from the burner is measured by comparison with the audiometer sound.

Types of Tests Run and Leading Results

In order to obtain results which could be compared it was important to plan and carry out all tests according to a definite procedure. Tests were run under a variety of conditions necessary to determine performance characteristics. These were designated by a series letter and the following presentation and discussion of leading results is included to illustrate the purpose of the various tests.

Series A. These were continuous tests with No. 1 oil (*i.e.,* fuel burned continuously at a uniform rate throughout test). Certain operating conditions were fixed in order to be reasonably sure that the burner was performing in essentially the same manner in each boiler. The CO_2 was standardized at 10 per cent, and with all burners except *OB 5* the minimum fuel burning rate

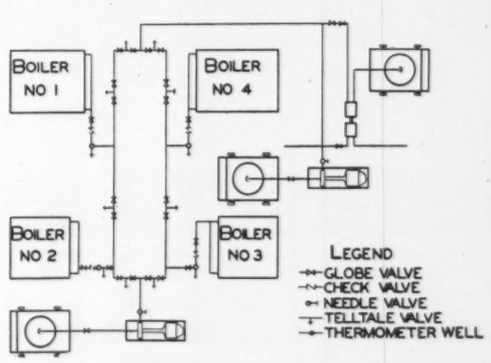


FIG 2—DIAGRAM OF FEEDWATER PIPING

was the lowest rate that could show 10 per cent ($\pm$ 0.4) CO_2 without apparent smoke.

The furnace draft was standardized at 0.02 in. of water and another test was run to determine the maximum rate of fuel burning possible with 10 per cent CO_2 and the draft of 0.02 in. Rates of oil burning exceeding

[1] Assistant Professor of Mechanical Engineering, Sheffield Scientific School, Yale University, New Haven, Conn.
[2] Graduate student and Assistant in Mechanical Engineering, Sheffield Scientific School, Yale University, New Haven, Conn.

this maximum were generally run if the rate of oil feed was capable of an increase. In this last test the furnace draft exceeded 0.02 in. water.

1. Fig. 3 shows maximum and minimum oil burning rates obtained within the 10 per cent CO_2 and 0.02 in. draft limitations imposed. It should be understood that a study of this type requires that some CO_2 percentage and furnace draft be chosen in order to be able to make valid comparisons. It should also be pointed out that 10 per cent CO_2 is a reasonable working standard. The results reported in this paper indicate the performance that could logically be expected under the usual operating conditions. The furnace draft of 0.02 in. of water may be considered low but it seems desirable as will be shown. Furthermore, it is obvious that there will be maximum and minimum fuel burning rates for whatever percentage of CO_2 and furnace draft is fixed upon. Both values will shift together in the same direction, however, so that the range or difference between maximum and minimum rates may stay much as shown by Fig. 3.

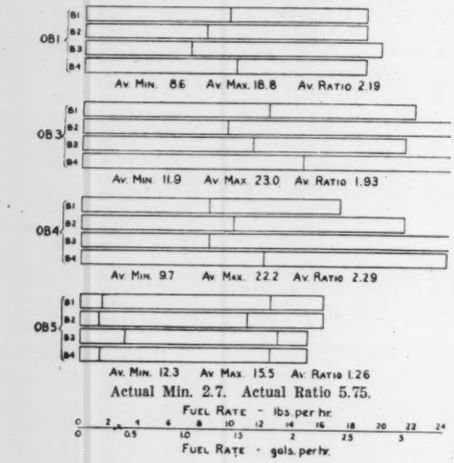

OB1
Av. Min. 8.6 Av. Max. 18.8 Av. Ratio 2.19
OB3
Av. Min. 11.9 Av. Max. 23.0 Av. Ratio 1.93
OB4
Av. Min. 9.7 Av. Max. 22.2 Av. Ratio 2.29
OB5
Av. Min. 12.3 Av. Max. 15.5 Av. Ratio 1.26
Actual Min. 2.7. Actual Ratio 5.75.
Fuel Rate - lbs. per hr.
Fuel Rate - gals. per hr.

FIG 3—MAXIMUM AND MINIMUM FUEL RATES OBTAINED WITH 10 PER CENT CO_2 AND 0.02 IN. WATER FURNACE DRAFT LIMITS

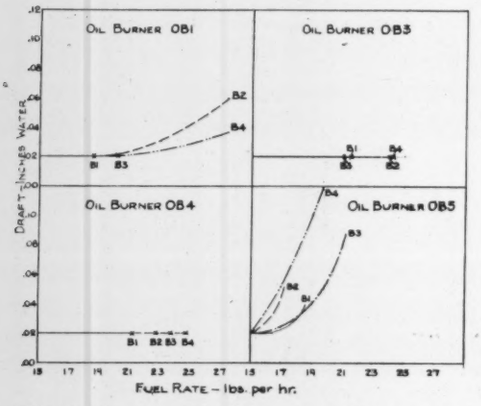


FIG. 4—GRAPHS SHOWING THE RELATION OF FUEL BURNING RATE AND FURNACE DRAFT FOR VARIOUS OIL BURNERS

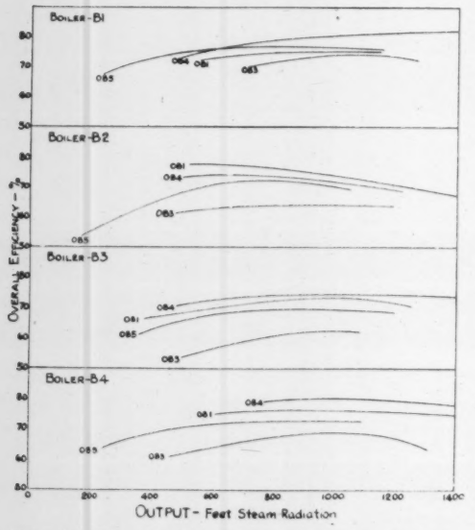


FIG. 5—OVERALL EFFICIENCY CURVES, GROUPED ACCORDING TO BOILERS

TABLE 1—DESCRIPTION OF LOW PRESSURE HEATING BOILERS TESTED*

	Designation of Boiler			
	B1	*B2*	*B3*	*B4*
Area of base, square feet.....	5.04	3.28	6.25	3.18
Water capacity to water line, lb.	345.5	303.5	236.3	747.0
Heating surface, square feet....	91	52.6	58.2	77
Total furnace volume, cubic feet	11.83	8.08	18.48	10.48
Average furnace width, inches...	22.0	22.0	30.0	21.0
Average furnace length, inches..	33.0	21.5	30.0	28.7
Average furnace height, inches..	28.2	29.5	35.5	30.0

* These boilers were requested for an output of 1000 sq ft of "radiation" and represented the nearest approach to this rating made by the manufacturer in the spring of 1929.

OB 5 is shown with two minimum values because this is a continuous operating type of burner. The attempt to go below the standard minimum conditions of 10 per cent CO_2 would cause the CO_2 of any of these burners to diminish. *OB 5* represents such a condition. The point of importance in this case is that an operating fuel rate ratio would have to be greater than the average value of about 2.0 shown for *OB 1, OB 3,* and *OB 4,* because, due to its continuous operation, this type must possess sufficient flexibility to meet exactly all fluctuations in heating loads caused by weather changes.

2. Fig. 4 shows another point in relation to maximum fuel burning rates and furnace draft. It will be noted in the cases of *OB 1* and *OB 5* that oil burning rates in excess of 20 and 15 lb per hour respectively, required the furnace draft to be increased. This merely indicates that the increased air required had to be assisted into the furnace by an increase of furnace draft. *OB 3* and *OB 4* did not reach such a limit during these tests.

All oil burners would probaby show curves like those of *OB 1* and *OB 5* if forced sufficiently unless the burner was limited by its ability to supply oil. *OB 3* was so limited in this case, but not *OB 4*. Practically, this curve suggests that oil burner manufacturers' ratings should be made at a point somewhere near the 0.02 in. of water furnace draft value. The fuel rates at maximum might be somewhat increased with no change in air, thus producing CO_2 greater than 10 per cent, but this increase would be comparatively small at best.

In addition to the maximum and minimum Series A tests just described, a number of similar tests were also

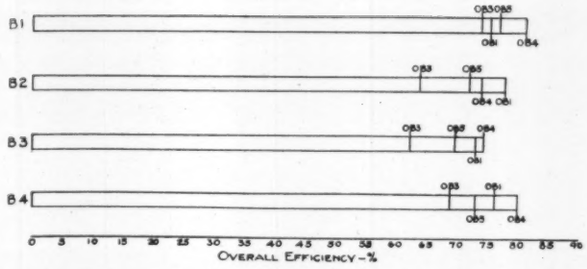


FIG. 6-*a*—MAXIMUM OVERALL EFFICIENCIES IRRESPECTIVE OF BOILER OUTPUT RATE, GROUPED ACCORDING TO BOILERS

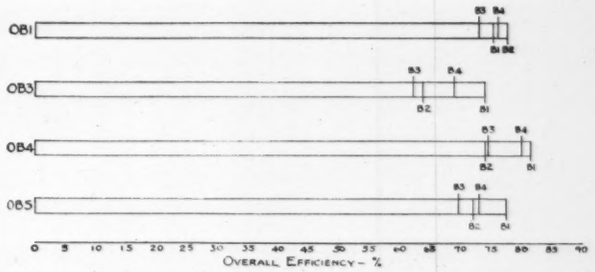


FIG. 6-*b*—MAXIMUM OVERALL EFFICIENCIES IRRESPECTIVE OF BOILER OUTPUT RATE, GROUPED ACCORDING TO BURNERS

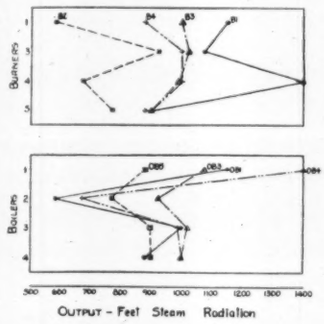


FIG. 7—CURVE SHOWING OUTPUTS AT WHICH MAXIMUM EFFICIENCIES OCCURRED, GROUPED ACCORDING TO OIL BURNERS AND TO BOILERS

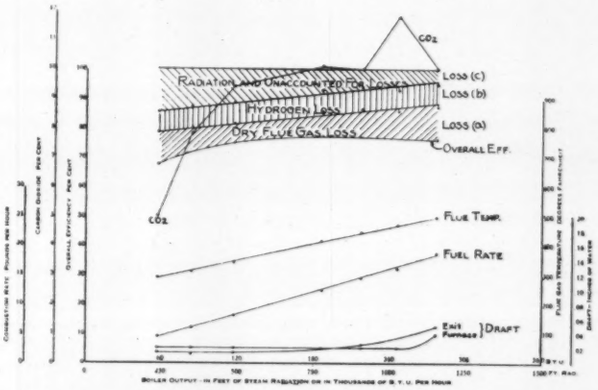


FIG. 8—PERFORMANCE CHART *OB 5—B1* HEAT BALANCE

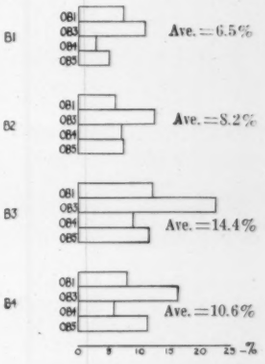


FIG. 9—PER CENT OF RADIATION AND UNACCOUNTED FOR LOSS (*c*) AT 17.5 LB-HOUR FUEL RATE, GROUPED ACCORDING TO BOILERS

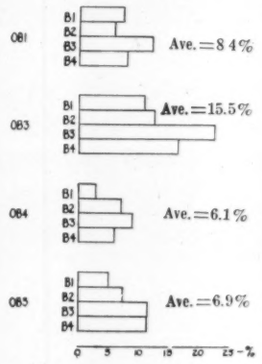


FIG. 10—PER CENT OF RADIATION AND UNACCOUNTED FOR LOSS (*c*) AT FUEL RATE OF 17.5 LB PER HOUR, GROUPED ACCORDING TO OIL BURNERS

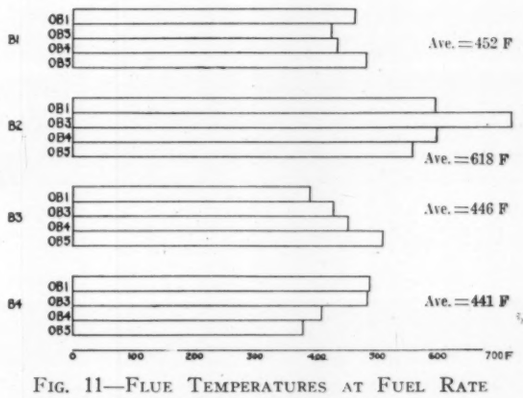


FIG. 11—FLUE TEMPERATURES AT FUEL RATE OF 17.5 LB PER HOUR, GROUPED ACCORDING TO BOILERS

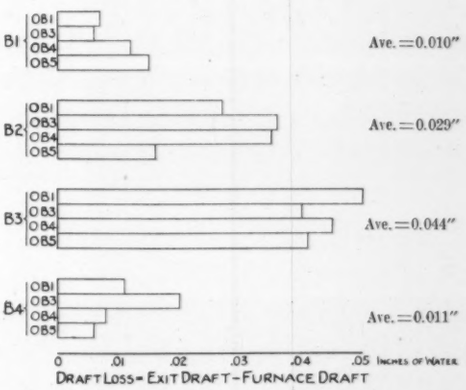


FIG. 12—DRAFT LOSS THROUGH BOILERS AT A FUEL RATE OF 17.5 LB PER HOUR

run at intermediate outputs, so that performance curves might be obtained. The complete results obtained from Series A tests are included in Figs. 29 to 44. Values obtained are overall efficiency, flue temperature, fuel rate, furnace draft, smoke hood draft and per cent CO_2. Other values have been obtained from the test data which will be enumerated.

3. Fig. 5 shows as the first fact that the boiler efficiency curve (*i.e.*, its shape) preserves its general character regardless of the oil burner. One would conclude from this evidence that the shape of an efficiency curve is largely a characteristic of the boiler. The fact that all the curves for one boiler do not coincide indicate that the oil burner does influence the value of the curve but not its character.

4. Fig. 6 (*a* and *b*) shows the maximum efficiencies obtained irrespective of boiler output at which said efficiency occurred. It is interesting to note that the range is from 64.5 per cent to 82.5 per cent for all boilers. That various oil burners produce different maximum values in the same boiler is also apparent, *OB 3* being the lowest in every case and *OB 4* being highest in three instances. The boilers in order of highest efficiency may be listed as *B 1, B 4, B 2* and *B 3*.

5. Fig. 7 shows the boiler outputs at which the maximum efficiencies in Fig. 6 were obtained. Maximum efficiencies for *B 1* occurred in all but one case at higher outputs than the other boilers. *B 3* and *B 4* show the same general results and *B 2* gave lowest outputs at maximum efficiency. Referring to Figs. 6 and 7, *OB 4*

—*B2* shows 80 per cent efficiency at 675 sq ft of equivalent direct "radiation" output and *OB 4—B 1* 82.5 per cent at about 1400 sq ft of "radiation" output. This illustrates what may be an important factor in practice, namely, that in considering the application of two boilers,

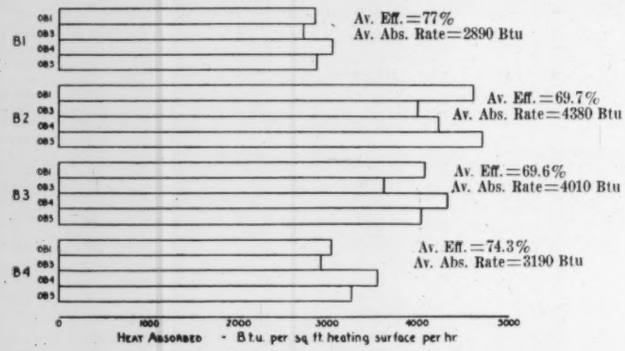


Fig. 13—Average Btu Absorbed Per Hour Per Square Foot of Boiler Heating Surface at a Fuel Rate of 17.5 Lb Per Hour

maximum efficiency should be stated in connection with output at which this efficiency is obtained.

6. Fig. 8 shows graphically an example of a complete heat balance. The losses are due principally to (*a*) loss in dry flue gas; (*b*) hydrogen or water vapor loss, and (*c*), radiation and unaccounted for losses. Loss (*a*) depends upon the weight of dry flue gas formed per pound of oil and the temperature of the flue gas. Loss (*b*) depends upon the hydrogen content of the fuel and the flue temperature. Loss (*c*) is found by difference in order to account for 100 per cent of the heat produced by the fuel.

It should be noted that loss (*a*) generally increases

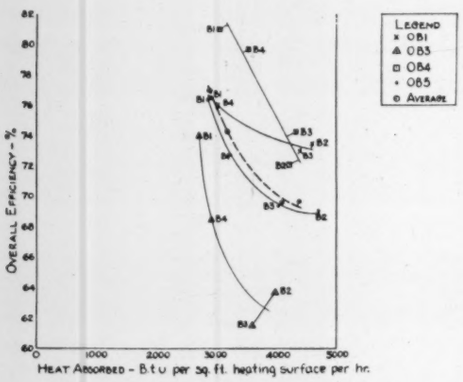


Fig. 14—Efficiency at Fuel Rate of 17.5 Lb Per Hour vs. Rate of Heat Absorption

with boiler output because flue temperature increases also. Loss (*b*) changes very little, increasing only slightly with increased boiler output. Loss (*c*) decreases with output because radiation losses remain substantially constant at all outputs, hence causing an actual percentage decrease as the boiler output increases.

An especially interesting fact about loss (*a*) in the case shown in Fig. 8 is that it is greater at 250 sq ft of "radiation" output than at 500 sq ft of "radiation," even though the flue temperature is lower at 250 sq ft of "radiation." The explanation is seen in the CO_2 curve which starts falling rapidly from about 500 sq ft of "radiation." A lower CO_2 means greater excess air and an increased weight of dry flue gas per pound of oil. Even though the gas temperature is lower at 250 sq ft of "radiation," the increase in weight of dry flue gas more than offsets it. Losses (*b*) and (*c*) are not greatly changed by excess air but big differences may occur in loss (*a*). This is the reason why oil and air adjustments should be measured by an Orsat to be sure that there is no unreasonable excess of air.

7. A further study of loss (*c*) Fig. 8 should furnish some clue as to why the same boiler may show changes in efficiency with various burners operating under standard series *A* conditions.

Fig. 9 shows these losses grouped according to boilers and based on identical fuel burning rate (*i.e.*, 17.5 lb per hour). A rate of 17.5 lb per hour was chosen as a reasonable one, giving an average boiler output of about

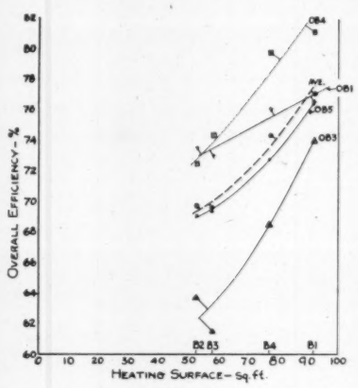


Fig. 15—Efficiency at Fuel Rate of 17.5 Lb Per Hour vs. Heating Surface

1,000 sq ft of "radiation." The most obvious fact is that *B 3* shows higher average losses than the other boilers. This can be accounted for by the fact that *B 3* has a high base, consequently a large exposed area, which is not water backed and as a result may allow higher radiation losses than the other boilers.

Fig. 10 shows the same values grouped according to oil burners. This shows that *OB 3* apparently causes higher radiation losses than the other burners. This can be accounted for by the fact that up to the present

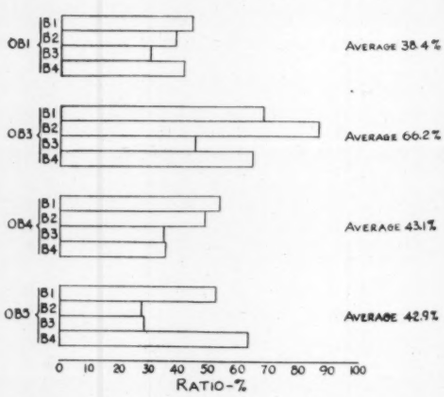


Fig. 16—Ratio of Furnace Volume *B* to Furnace Volume *A* (Furnace Volume *A*, Before Installation of Burner; Furnace Volume *B*, After Installation of Burner)

time the results obtained with this burner indicate that it heats the floor and ashpit sidewalls more than the other types and therefore causes higher radiation losses. It is evident that boiler and oil burner both may influence this value when all variations in design are considered.

8. Fig. 11 shows flue temperatures grouped according to boilers all at a fuel rate of 17.5 lb per hour. The temperatures vary from 380 to 720 F and it is obvious that temperatures for *B 2* average higher than the other boilers. Aside from this fact there is nothing else shown of a conclusive nature.

9. Fig. 12 shows a boiler characteristic which is the draft loss through the boiler for a fuel burning rate of 17.5 lb per hour. The values for one boiler and various oil burners are not the same in spite of the fact that the weight of flue gas per pound of oil is the same in all cases. It is apparent that draft losses vary from 0.005 to 0.05 in. of water, which is a wide variation. That the boilers may be classified from high to low as to amount of draft required is evident as follows: *B 3, B 2, B 4, B 1.* Boilers of this type operating on comparatively low chimneys should not indicate too high a draft loss but there is some evidence that the resistance need not be reduced to a minimum either.

Table 2—Heat Absorption Rate at Maximum Efficiency in Btu Per Square Foot of Heating Surface Per Hour

	OB 1	*OB 3*	*OB 4*	*OB 5*	Average
B 1	3032	2835	3692	2329	2972
B 2	2674	4225	3080	3536	3379
B 3	4124	4206	4082	3711	4031
B 4	2743	3117	3117	2805	2945
Average	3143	3596	3493	3095	

10. Another feature of interest shown by series *A* tests is the average heat absorbed per square foot of boiler heating surface for fuel rates of 17.5 lb per hour in all cases. Fig. 13 shows rates ranging from 2,700 to 4,700 Btu absorbed per square foot of heating surface per hour. The boiler showing the highest value is *B 2,* and in descending order, *B 3, B 4* and *B 1.* The average overall efficiency of all four burners in each boiler at the same fuel rate of 17.5 lb per hour is as follows:

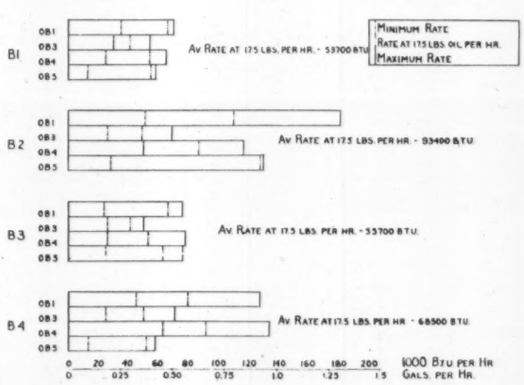

B1
B2
B3
B4
OB1
OB3
OB4
OB5
Av Rate at 17.5 lbs. per hr. - 53700 BTU
Av Rate at 17.5 lbs. per hr. - 93400 BTU
Av Rate at 17.5 lbs. per hr. - 53700 BTU
Av Rate at 17.5 lbs. per hr. - 68500 BTU
Minimum Rate
Rate at 17.5 lbs. oil per hr.
Maximum Rate
0 20 40 60 80 100 120 140 160 180 200
0 .025 .050 .075 .10 .125 .15
1000 Btu per Hr
Gals. per Hr.

Fig. 17—Maximum and Minimum Heat Liberation Rates in Thousands of Btu Per Cubic Foot Per Hour of Furnace *B*

B 2—69.7 per cent, *B 3*—69.6 per cent, *B 4*—74.3 per cent and *B 1*—77.0 per cent.

Fig. 14 shows clearly that higher efficiencies are accompanied by lower heat absorption rates but that these values vary with the burner. It is evidently a characteristic of *OB 4* that it can obtain heat absorption rates at better efficiency than other types of burners. *OB 1* has similar characteristics at high heat transfer rates. Both *OB 1* and *OB 4* have a similar type of flame.

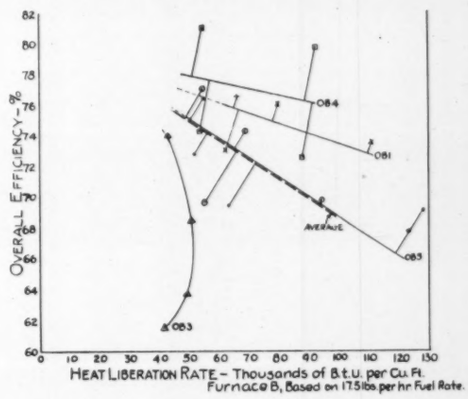


Fig. 18—Efficiency at Fuel Rate of 17.5 Lb Per Hour vs. Heat Liberation Rate

Table 2 shows the heat absorption rate per square foot of heating surface per hour at the point of maximum efficiency. The minimum rate is 2,329 Btu with *OB 5—B 1* and the maximum is 4,225 Btu with *OB 3—B 2.* It is interesting to note that the average rate of each of the burners in the different boilers is between 3,000 and 3,600 Btu. The average rate of each boiler with the different burners is not within as short a range as the average rate of each burner.

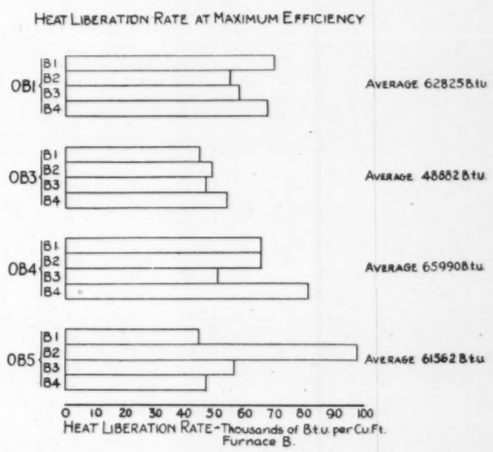


Fig. 19—Heat Liberation Rate at Maximum Efficiency

11. Fig. 15 represents the overall efficiency obtained at a fuel rate of 17.5 lb per hour compared to the heating surfaces in the respective boilers. This shows that for a given rate of heat generation the most heating surface will absorb the most heat. It is possible for the design of the surface to be such that two boilers of equal surface will not absorb an equal quantity of heat. The average difference between *B 2* and *B 3* is negligible but loss (*c*) Fig. 9 gives a clue in explanation thereof.

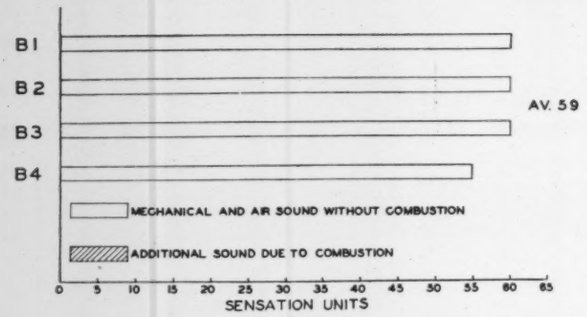


FIG. 20—SOUND EMITTED BY BURNER *OB 1*, SERIES *G*

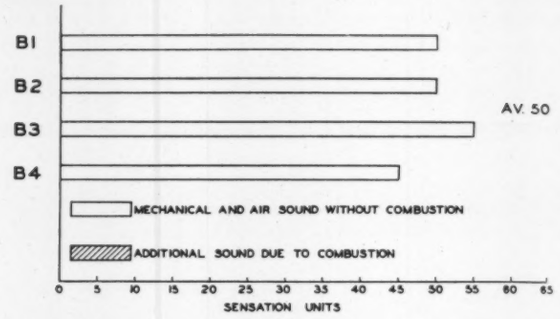


FIG. 21—SOUND EMITTED BY BURNER *OB 3*, SERIES *G*

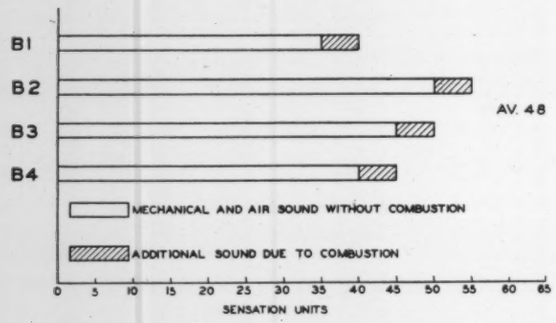


FIG. 22—SOUND EMITTED BY BURNER *OB 4*, SERIES *G*

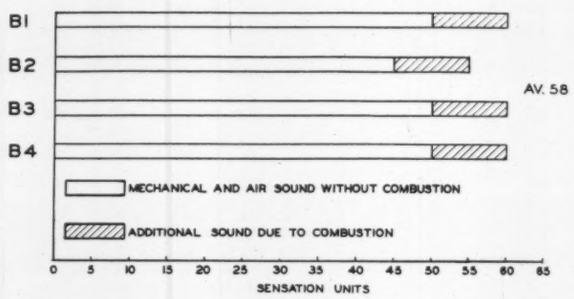


FIG. 23—SOUND EMITTED BY BURNER *OB 5*, SERIES *G*

12. *Heat Liberation Rates in Furnace*: Various values have been submitted for the desirable rate of burning oil per cubic foot of furnace volume. First of all there are two furnace volumes—(*a*) the space available for fuel burning within the boiler as supplied by manufacturer—called furnace *A*—and (*b*) the available fuel burning space after the burner has been installed—called furnace *B*. It is evident that various types of burners will occupy different proportions of furnace *A* according to type.

Fig. 16 illustrates the per cent of furnace *A* left over after the burner installation, showing that furnace *B* may be from 27.5 to 86.5 per cent of furnace *A*. This is a considerable variation. *OB 3* is shown as having a greater per cent of furnace *A* for combustion purposes; the others are about the same. This is caused by the nature of the furnace bricking and the proportion of burner actually placed within the furnace. The true measure of heat liberation rate in thousands of Btu per hour per cubic foot of furnace volume should naturally be based on the volume actually available which is furnace *B*. See A. S. H. V. E. Proposed Code for Testing Steam Heating Boilers Burning Oil Fuel for definition of furnaces *A* and *B*.

Fig. 17 shows minimum, 17.5 lb per hour and maximum rates per cubic foot of furnace *B*. The lowest rate shown is 12,500 and the highest rate 181,000 Btu per cubic foot of furnace *B* per hour. At 17.5 lb per hour the minimum is 40,800 and maximum is 127,500 Btu. *B 1* shows the lowest average rate at 17.5 lb per hour at 53,700 Btu and *B 2* the highest at 93,400 which is equivalent to 2.73 to 4.74 lb of oil per cubic foot of furnace *B* per hour. *OB 3* shows lowest average for reason by Fig. 16.

The general relationship between heat liberation rate and efficiencies is shown by Fig. 18. Both values are those obtained at a fuel rate of 17.5 lb per hour. At a glance it appears that the curves on the figure are far off from the points. This impression will be corrected if the fact is noted that the vertical scale is magnified and that the maximum deviation from any of the curves is just a trifle beyond that allowed for experimental error. The significant facts about these curves are their negative slope to the right. Also, it should be noted that the curves for *OB 4*, *OB 1*, and *OB 5*, in this order, lie above the average curve and the curve for *OB 3* is below this curve.

The curve for *OB 3* has a form and direction different from the three other curves. The *OB 3* curve shows an increase in efficiency accompanied by a decrease in heat liberation rate up to a certain point. Beyond this point a decrease in heat liberation rate is accompanied by a decrease in efficiency.

In order to ascertain whether or not there is a desirable heat liberation rate per cubic foot of furnace volume *B*, the heat liberation rates were determined at points of maximum efficiency. Fig. 19 shows that heat liberation rates at maximum efficiency may vary from 44,600 to 97,400 Btu per hour.

Due to variations in furnace *B* and in burner design it is not clear that any specific value can be chosen as a desirable rate of fuel burning. Any rate would seem reasonable which produces complete combustion and gives an amount of heat that the boiler can efficiently absorb. Heat liberation rate cannot be divorced from the heat absorbing possibilities of the boiler.

13. *Sound measurements*: Figs. 20 to 23 show the sound emitted classified according to burners, measurements being made by the method previously described. Two measurements were made, one with and one without combustion to detect whether or not combustion sounds were present. *OB 1* and *OB 3* were found to be

free from combustion sounds; *OB 4* showing an increase of 5 sensation units and *OB 5* of 10 sensation units. *OB 1* and *OB 5* gave the highest sound measurements. Average values indicate the following desirability of the burners as to sound: *OB 4, OB 3, OB 5, OB 1.*

The discussion of an item of this sort is a delicate matter because of variations in personal sensibilities and because of little background based on facts. It is also realized that the sound measured in a laboratory may not indicate what an actual installation may do on account of sound and vibration transmissions due to piping, building construction, chimney, etc. It is reasonably certain, however, that all other things being equal, the burner producing the smallest sound at the boiler will be generally the most satisfactory.

It seems evident that the sound impression is not strictly proportional to the measurements shown. Sounds of 40 and 50 sensation units do not seem unpleasant, whereas 55 and 60 seem quite the contrary. It is likewise evident that most of the sound is due to mechanical and air conditions. Fan air inlets may be profitably studied in a similar manner to the attempts of automotive engineers to reduce the sound of air entering carburetors. The effect of boiler types on sound is not very conclusive although *B 4* seems better in two cases. No definite reason can be given for this, however.

The sound measurements were made under standard series *A* conditions at about 1,000 sq ft of "radiation" output. It is generally known that variations in fuel-air proportions may in some cases produce variations in sound, but this special matter was not included in these tests.

14. *Power requirements*: The power required by various burners is given in Table 3. These are average values for all boilers at the different fuel rates at which series *A* was run.

TABLE 3

BURNER	POWER REQUIRED (WATTS)
OB 1	58.2
OB 3	181.4
OB 4	85.8
OB 5	75.5

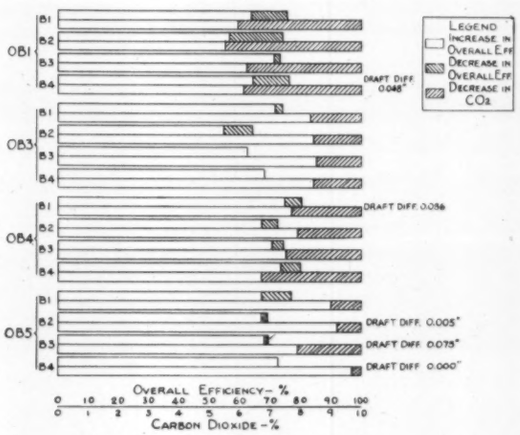


FIG. 24—COMPARISON OF EFFICIENCIES AND CO_2 OF SERIES *A* AND *C* AT THE SAME FUEL RATE. DRAFT DIFFERENCE 0.05 IN. WATER EXCEPT WHERE NOTED

Series B: Series *B* is a set of tests similar to series *A*, except that No. 3 oil was used instead of No. 1. This was done with burners which the manufacturer recommends for oils No. 1, 2 and 3. *OB 1* and *OB 3* were such burners.

15. So far as efficiency is concerned, *OB 1* showed small differences in amount or shape of curve. *OB 3* showed up on the average somewhat better with the No. 3 oil and in two cases the shape of the curve was different, the maximum efficiency occurring at lower outputs. In 7 out of 8 combinations of boilers and burners the efficiency was somewhat lower with No. 3 than with No. 1 oil at the highest outputs. The overall differences otherwise are not considered important and no graphs will be shown.

It must be clearly understood that the foregoing comments refer only to combustion and not to other features of operation, such as ignition and other factors which may show differences caused by various grades of oil. There are nowhere in this report any facts or opinions relative to those features necessary for safety and satisfactory automatic operation. These tests are confined to the heat-generating or oil-burning characteristics of the burner and the factors influencing these characteristics. The concern of this study is not a question of how the oil ignites but how it burns.

Series C: Series *C* is one test designed for one pur-

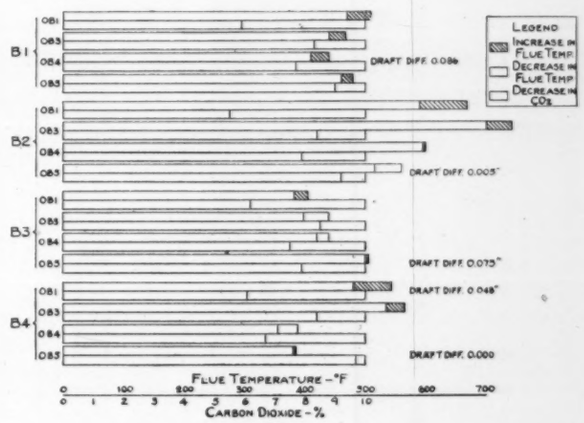


FIG. 25—COMPARISON OF FLUE TEMPERATURES OF SERIES *A* AND *C* AT THE SAME FUEL RATE. DRAFT DIFFERENCE 0.05 IN. WATER EXCEPT WHERE NOTED

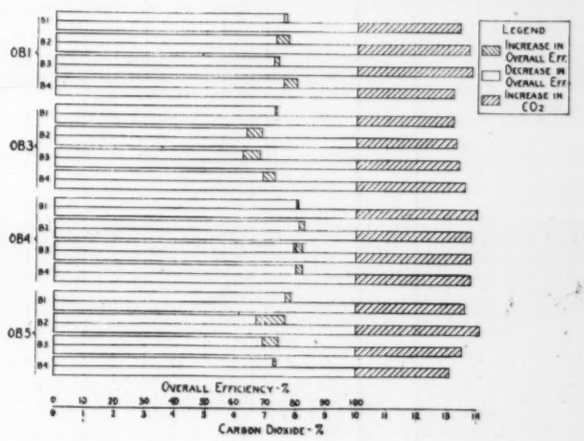


FIG. 26—COMPARISON OF EFFICIENCIES AND CO_2 OF SERIES *A* AND *D* AT THE SAME FUEL RATE

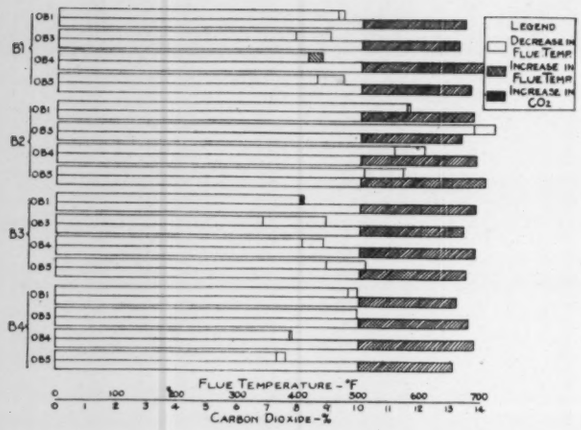


FIG. 27—COMPARISON OF FLUE TEMPERATURES OF SERIES *A* AND *D* AT THE SAME FUEL RATE

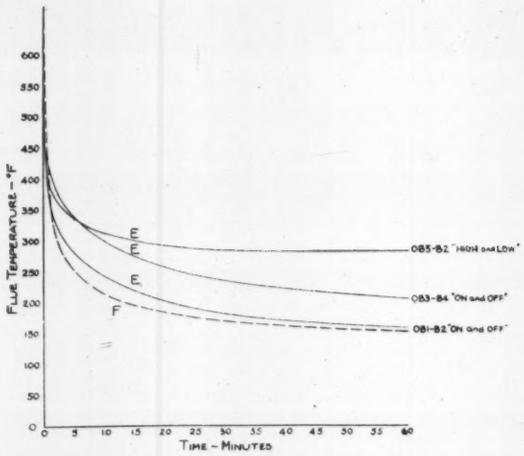


FIG. 28—CHANGE IN FLUE TEMPERATURE DURING AN *Off* PERIOD PLOTTED AGAINST TIME. (*E* REPRESENTS 0.02 IN. DRAFT SETTING AND *F* REPRESENTS 0.07 IN. DRAFT SETTING)

pose and that purpose is merely to indicate the effect of a draft change. The conditions of the test are similar to a series *A* test, of a corresponding fuel rate, giving an output of about 1000 sq ft of "radiation," with only one difference and that is the change in furnace draft from 0.02 to 0.07 in. of water. Does this affect combustion efficiency and flue temperature?

16. Fig. 24 shows the effect on combustion and efficiency for a 0.05-in. water change in furnace draft. *OB 5, OB 4* and *OB 1* show a few cases where the draft difference was other than 0.05 in. and are so marked. The upper of each pair of bars is efficiency, that portion shaded showing the decrease caused by draft increase. The lower of each pair of bars represents the standard series *A* 10 per cent CO_2 value, the shaded portion showing the decrease in CO_2.

The decrease in efficiency may be from zero to 17.7 per cent and CO_2 from 1 to 4.5 per cent, not including *OB 5* and those of *OB 4* and *OB 1* in the same category as *OB 5*, for the reason already stated. It should be noted that *OB 1* is particularly influenced by draft change, that *OB 4* is moderately influenced, and that *OB 3* and *OB 5* are somewhat variable.

17. Fig. 25 shows the effect on flue gas temperatures. The upper bar of each pair shows flue temperature, the shaded portion shows the increase in temperature due to increase in draft, and the unshaded shows the decrease. The results are not uniform, but the majority of cases indicate that flue temperatures may increase when the excess air increases. This aggravates the loss because not only is the weight of dry flue gas increased, but its temperature also—a double penalty, so to speak.

There are, of course, some cases where the reverse is indicated. *OB 1,* which showed the biggest drop in CO_2, shows the largest average flue temperature increase. It is not certain that the result (*i.e.*, whether flue temperature increases or decreases) can be foretold in all cases. It is thought that a boiler of small heating surface may show a decrease and vice versa an increase. Note that *B 1* shows an increase in all cases.

18. Series *C* shows obviously that furnace draft is one of the important factors in burner operation. It must be adjusted at the same time that the oil and air are being set and it must be kept at a constant value just as oil and air should.

Series D: Series *D* is again a single test for a single purpose. All conditions of a series *A* test were obtained at an output of about 1000 sq ft of "radiation" with one exception. This exception is CO_2 and it is made as high as possible without visible smoke, according to the best judgment of the tester.

19. Fig. 26 shows increased CO_2 as high as 14.1 per cent and increased efficiencies in all but one case. It will

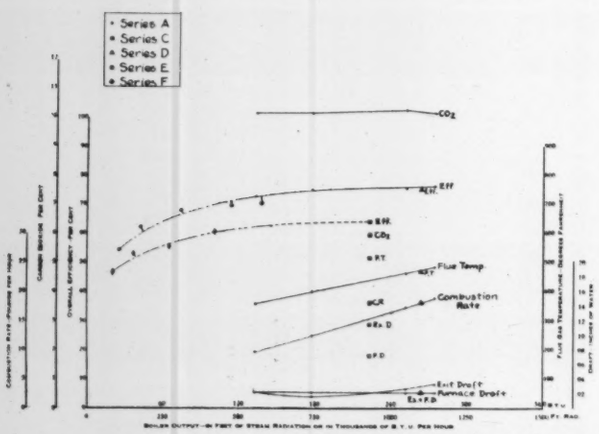


FIG. 29—PERFORMANCE CHART *OB 1—B 1*

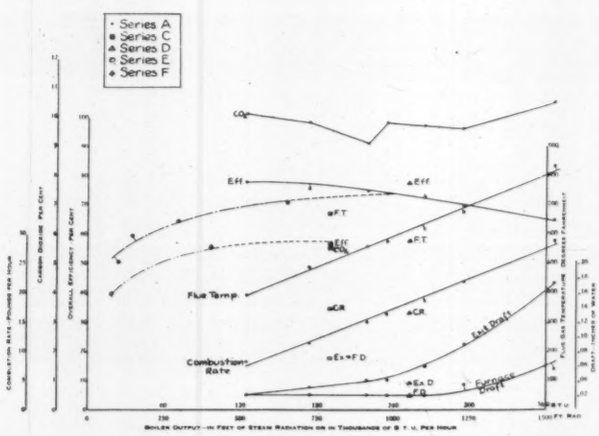


FIG. 30—PERFORMANCE CHART *OB 1—B 2*

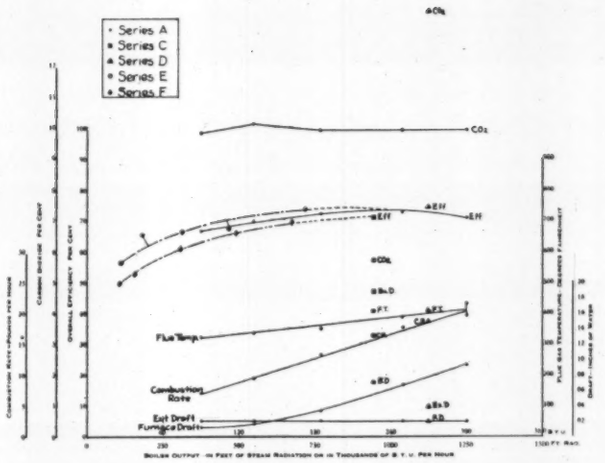


FIG. 31—PERFORMANCE CHART *OB 1—B 3*

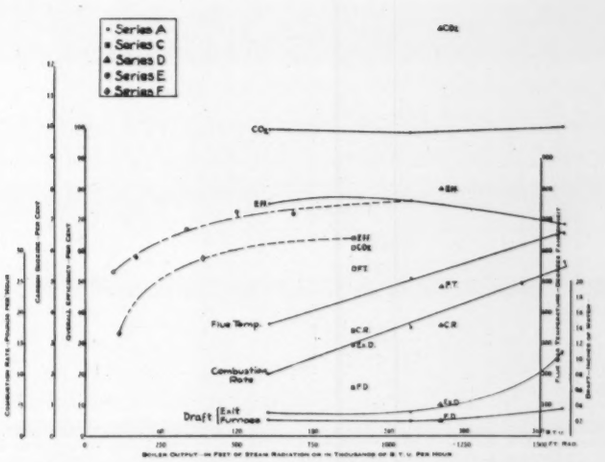


FIG. 32—PERFORMANCE CHART *OB 1—B 4*

be noted that the increase in efficiency for CO_2 above 10 per cent is not as great in proportion as the decrease in efficiency with decrease in CO_2 (See Fig. 24). It is evident from a practical standpoint that even though maximum CO_2 is ideal, there is a decreasing incentive to reach this ideal. A CO_2 value of 10 per cent was originally chosen for this work as a reasonable value to use. Furthermore, it is by no means certain that for everyday use a very high CO_2 might not be productive of other difficulties which would more than offset the gain in efficiency.

20. Fig. *27* compared to Fig. 25 shows that with an increase in CO_2, flue gas temperatures generally show a decrease. This would be expected from a study of Fig. 25. It probably is true that combustion temperatures are higher in series *D* tests than in those of series *C*. Why should the flue temperatures show the reverse effect? The answer evidently is in the effect of combustion temperature on heat radiation rate in the furnace. The radiation rate does not increase directly with temperature, but as a power of the temperature commonly considered the fourth power. For example, a 200-deg. increase in combustion temperature may only be a 10 per cent increase, but this change in heat radiation rate may be 30 per cent. It is evident that the more heat absorbed by the boiler furnace the less there will be for the heating surface in the flues to absorb.

Series E: Series *E* represents the intermittent type of test mentioned in the proposed test code. A standard series *A* condition at about 1000 sq ft of steam "radiation" output is obtained and then various combinations of *on* and *off* periods are utilized to obtain varying average outputs. By this scheme an efficiency curve may be made.

21. Figs. 29 to 44 show the results, but it should be emphasized that the point of highest efficiency is about where the burner operates continuously. Due to losses during the *off* periods the overall efficiency will decrease more and more as the *off* period durations are longer and longer. The characteristic efficiency curve is the same shape for all boilers in contrast to the showing for series *A* (Fig. 5).

It is interesting to note that as a general fact the series *E* and *A* curves will not be greatly different if, when the burner is on, it is operating at or near the maximum efficiency of the boiler. Although a high operating efficiency is desirable, it is evident that average efficiencies will be less. The higher the *on* efficiency the higher will be the average, however. *OB 3—B 3* (Fig. 35) seems to contradict this, but it is probably an undetected error of some sort.

22. It should be noted that *OB 5* may operate continuously by a graduated fire or by a *high* and *low* type of operation. Except for *OB 5—B 1* (Fig. 41), it may be said that either method will produce about the same result. *OB 5* as supplied gave 13 per cent CO_2 and it

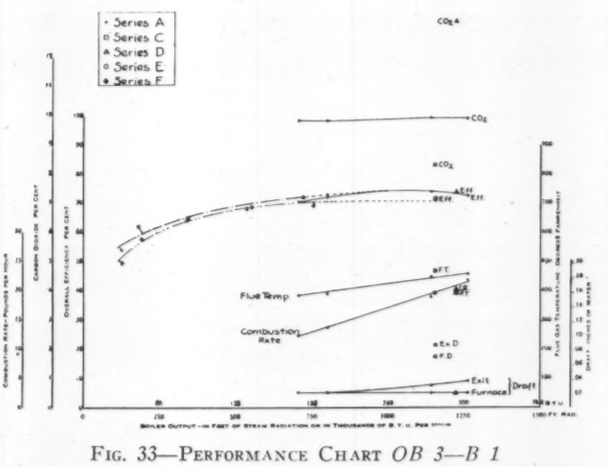


FIG. 33—PERFORMANCE CHART *OB 3—B 1*

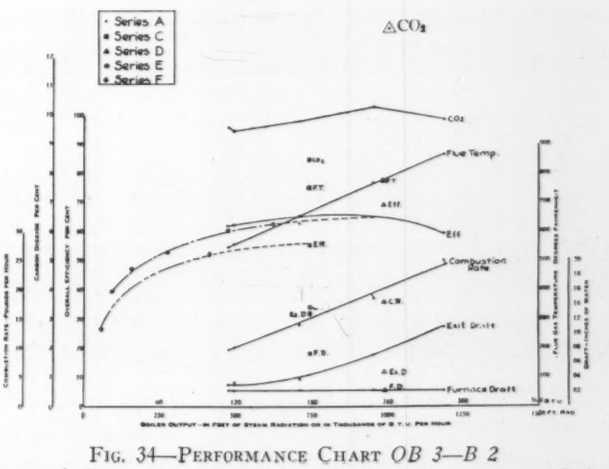


FIG. 34—PERFORMANCE CHART *OB 3—B 2*

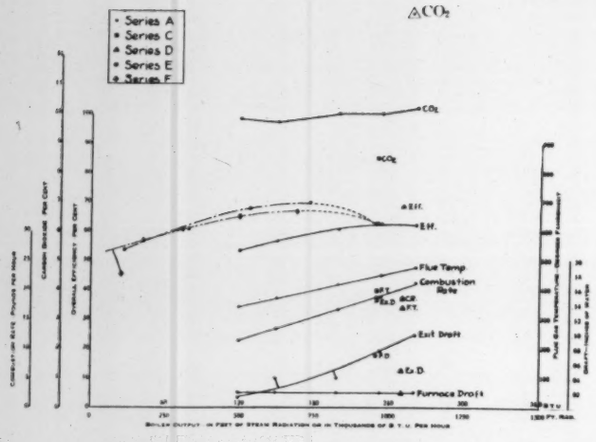


FIG. 35—PERFORMANCE CHART *OB 3—B 3*

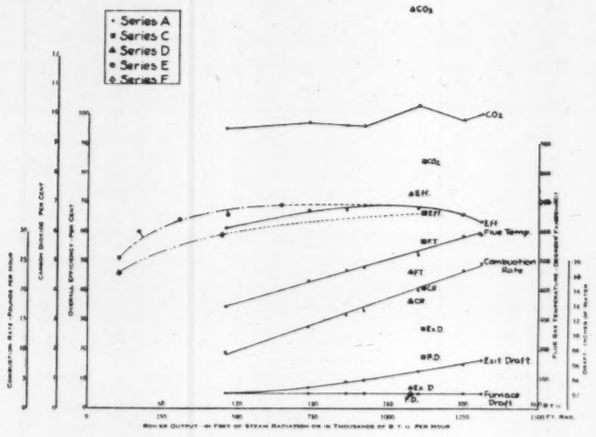


FIG. 36—PERFORMANCE CHART *OB 3—B 4*

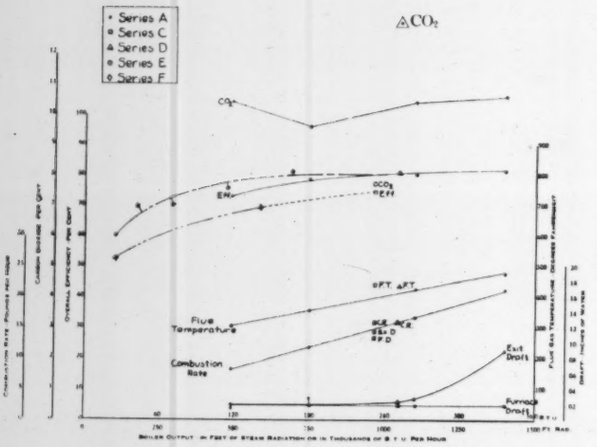


FIG. 37—PERFORMANCE CHART *OB 4—B 1*

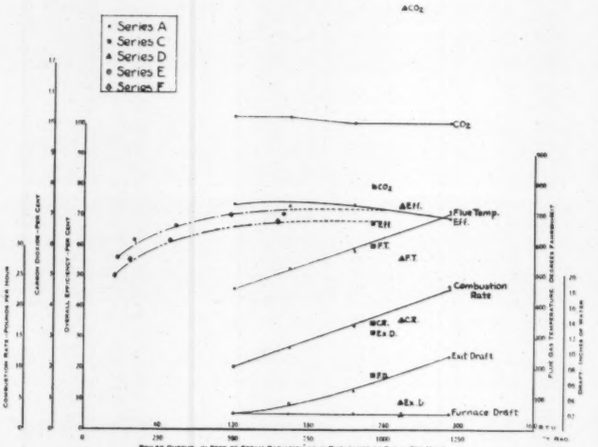


FIG. 38—PERFORMANCE CHART *OB 4—B 2*

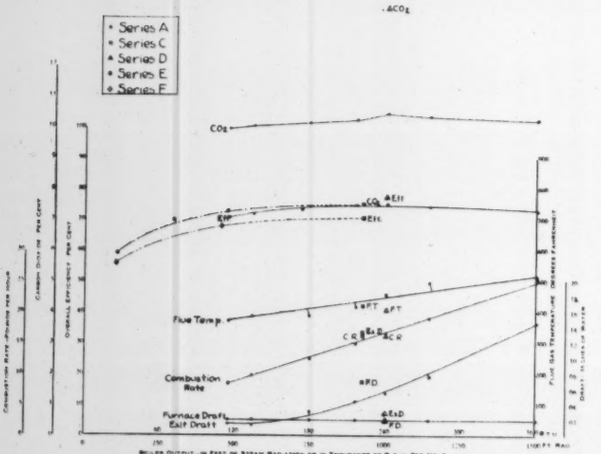


FIG. 39—PERFORMANCE CHART *OB 4—B 3*

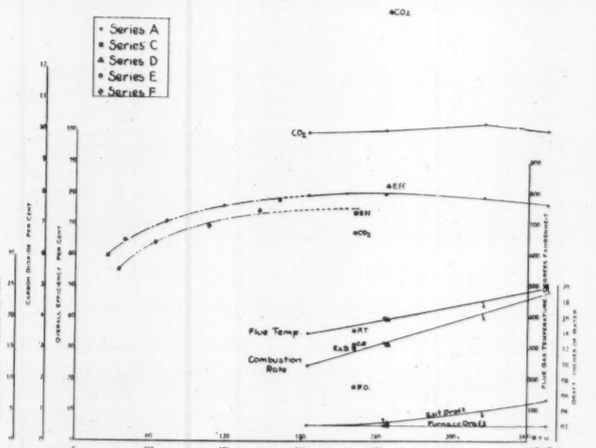


FIG. 40—PERFORMANCE CHART *OB 4—B 4*

was necessary to modify certain air ports to decrease the CO_2. It is thought that a *high* and *low* with a 13 per cent CO_2 would improve the efficiency considerably.

It is of further interest to note that the CO_2 drops to low values at small outputs, which undoubtedly is one of the causes of the reduced efficiency, but in spite of this the performance from an efficiency standpoint is not greatly different from the *on* and *off* types of burners.

23. Fig. 28 shows cooling rate curves as typical of what happens as soon as the oil burner shuts off. Note that *OB 5* goes on *low* operation and that the curve flattens out at about 270 F.

Series F: Series *F* is similar to series *E* except that the furnace draft is increased to 0.07 in water and intermittent tests run again.

24. The results shown in Figs. 29 to 44 merely substantiate the results of series *C* tests as to loss of efficiency. Series *F* efficiency curves are similar to those of

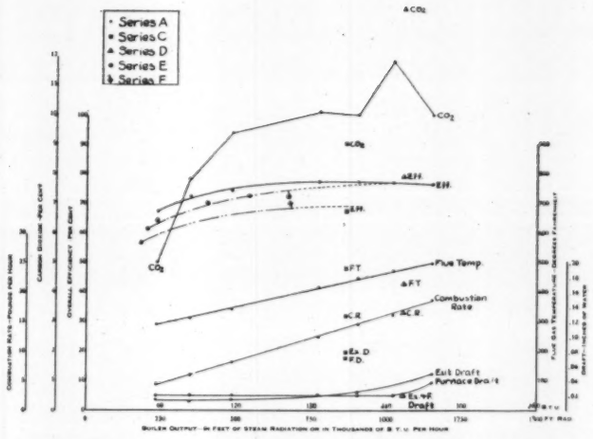


FIG. 41—PERFORMANCE CHART *OB 5—B 1*

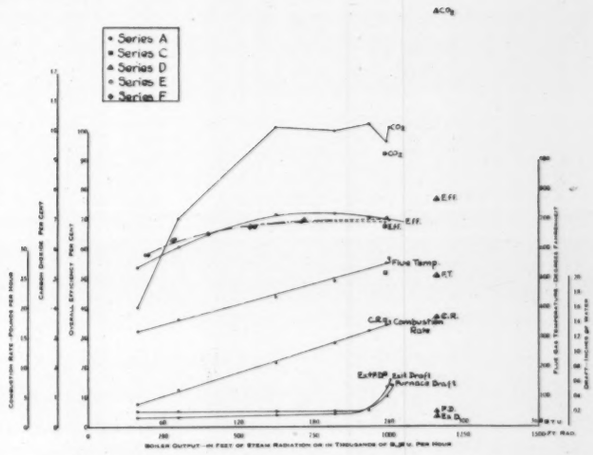


FIG. 42—PERFORMANCE CHART *OB 5—B 2*

series *E* but more or less lower in value. It is evident that a dependable method of preventing the flow of air into and through the boiler during the *off* periods would do much to conserve heat. The presence of a fan does render a burner less sensitive to draft fluctuations and a high-pressure, high air velocity fan will afford greater resistance to air flow during *off* periods. Fig. 28 shows the difference in cooling rate time for *OB 1—B 2* for series *E* and *F*.

25. In all of the intermittent tests the *on* or *high* was for 30 min. No other combinations of *on* periods were tried. Thirty minutes were chosen because it was believed that this was fairly typical for residences thermostatically operated. If pressure regulation is used the number of *off* periods may be greater in number per 24 hrs. If this is so the average operating efficiency is likely to be lower. It is hoped to investigate this point later.

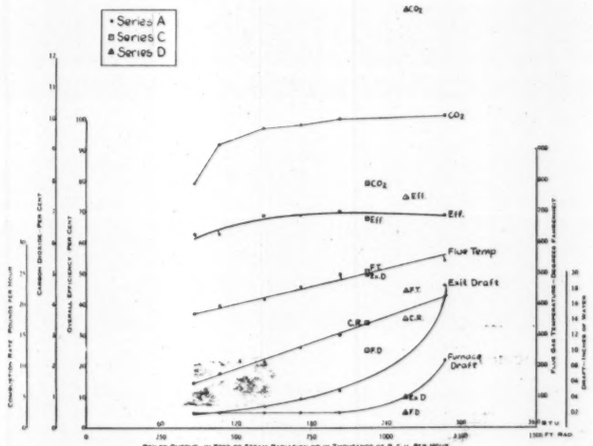


FIG. 43—PERFORMANCE CHART *OB 5—B 3*

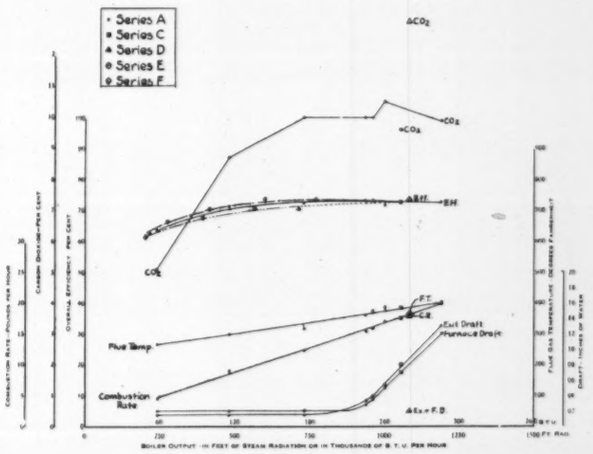


FIG. 44—PERFORMANCE CHART *OB 5—B 4*

26. The method of regulation and the type of burner (*i.e.*, continuous and intermittent) should receive future study as to sizing and adjusting for fuel rate in relation to the heating load. Pick-up load would be the important factor to consider. The characteristics of the boiler should likewise receive study in the same connection.

Series G: Series *G* were the sound tests reported in Section 13 in connection with series *A* results.

Summary

(1) The general shape of the boiler efficiency curves remains the same with various oil burners. (2) The efficiency values for a given boiler are not necessarily equal for various oil burners even when the oil burners are operating in practically the same manner in each case. (3) The radiation and unaccounted losses average higher for some boilers than others. (4) Some burners also show higher radiation and heat losses than others. (5) All other things being equal an increase in draft: (*a*) lowers boiler efficiency (*b*) lowers CO_2 (*c*) increases excess air and (*d*) increases flue temperatures. (6) All other things being equal a decrease in excess air when CO_2 is 10 per cent; (*a*) increases CO_2 (*b*) increases efficiencies moderately and (*c*) decreases flue temperatures. (7) Efficiency curves on intermittent type of operation all show the same characteristic shape. (8) Intermittent operation at different drafts shows that some burners are more sensitive to draft fluctuations than others. Item (5) also indicates the same characteristic. (9) Efficiencies tend to rise with a decrease in heat absorption rate of boiler per square foot of heating surface although some types of burners will give good results at higher rates than others. (10) Increase in boiler heating surface for the same fuel burning rates will show higher efficiencies. (11) Heat liberation rates in furnaces vary quite widely, averaging higher for some oil burners than others. There seems to be no such thing as a desirable fuel burning rate per cubic foot of furnace volume for oil burners in general. (12) On sound tests some burners make measurable combustion sounds but it does not necessarily follow that such a burner may not be more quiet than one giving no combustion sounds at all.

Friction Heads in One-Inch Standard Cast-Iron Tees

By F. E. Giesecke[1] (*MEMBER*) and W. H. Badgett[2] (*NON-MEMBER*)
College Station, Texas

This paper is the result of research conducted at Agricultural and Mechanical College of Texas in cooperation with the A. S. H. V. E. Research Laboratory.

THE object of this investigation was to determine the friction heads in a one-inch standard cast-iron tee when 70 F water was flowing through in the several possible directions.

Apparatus

The apparatus used for the investigation is shown in Figs. 1 and 2. In Fig. 1, *T* represents the tee under investigation and *AT*, *TB* and *TC* represent sections of standard black-iron pipe each about 5 ft long. The three pipe sections were cut from the same piece of pipe in order that the diameters of the three sections might be as nearly equal as possible. *R1*, *R2*, *R3*, *R4* and *R5* represent piezometer rings, to determine the pressure heads at their respective locations. *G1*, *G2*, *G3*, *G4* and *G5* represent glass tubes for obtaining the differences between the pressure heads at the several piezometer rings. *D1* and *D2* represent ½-in. circular diaphragm orifices for determining the quantity of water flowing in the respective pipe sections. *V1* and *V2* represent gate valves for regulating the flow of water through the two respective pipe sections.

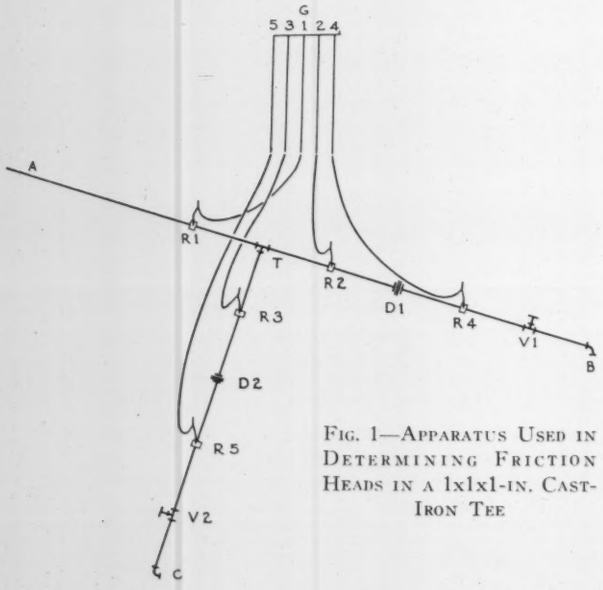


FIG. 1—APPARATUS USED IN DETERMINING FRICTION HEADS IN A 1x1x1-IN. CAST-IRON TEE

During a test, water entered the apparatus at *A* (Fig. 1) and was discharged at *B* and at *C*, the relative quantities flowing in the two directions being regulated so that the quantity flowing out at *B* varied from zero to 100 per cent, while the quantity flowing out at *C* varied from 100 per cent to zero. Provisions were made by which the water flowing through the apparatus was maintained at a mean temperature of 70 F with a maximum variation from that mean of less than one degree.

Fig. 2 shows a photograph of the apparatus while water was flowing so that 82 per cent was being discharged at *B* and 18 per cent at *C*. For this particular test a small quantity of red dye was added to the water to color it so that the elevations of the water in the glass tubes (*G*) could easily be seen in the photograph.

Calculation of Friction and Velocity Heads

The friction head for this case was calculated as follows: The difference in the elevations of the water in *G1* and *G3* (0.195 in.) was the loss in pressure head while the water was flowing from *R1* to *R3*. The difference in the elevations of the water in *G3* and *G5* (0.435 in.) was the loss of head while the water was flowing from *R3* to *R5* and through the orifice *D2*. From the calibration of this orifice, referred to in the paper Loss of Head in Submerged Orifices, by F. E. Giesecke,[3] it was found that the velocity of the water in the pipe section *TC* was 3.4 in. per second. However, in order to check this result, the water flowing through the pipe section *TC* was weighed, and the velocity determined from this weight was found to be 3.33 in. per second. In a similar manner, the velocity of the water

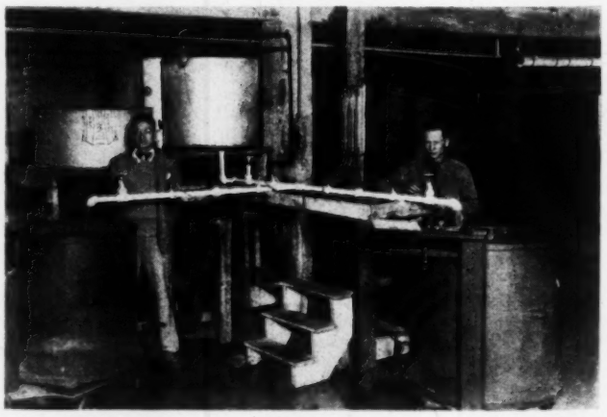


FIG. 2—APPARATUS USED IN DETERMINING FRICTION HEADS IN A 1x1x1-IN. CAST-IRON TEE

[1] Director, Texas Engineering Experiment Station, Agricultural and Mechanical College of Texas, College Station, Texas.
[2] Research Assistant, Texas Engineering Experiment Station, Agricultural and Mechanical College of Texas, College Station, Texas.
[3] See JOURNAL of The Society (Heating, Piping and Air Conditioning), July, 1930.

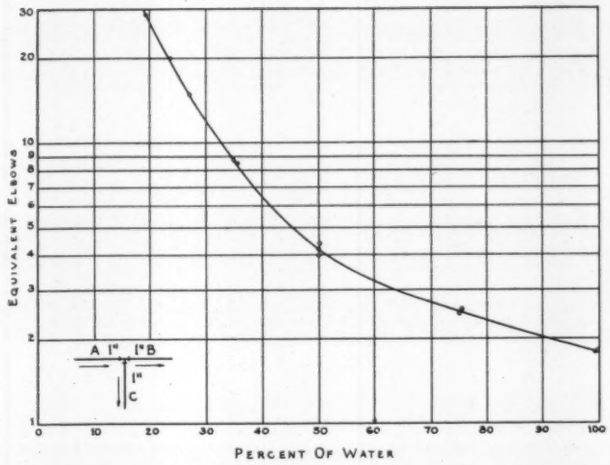


FIG. 3—THE FRICTION HEAD IN A 1x1x1-IN. CAST-IRON TEE WHEN WATER ENTERS AT *A* AND A PORTION IS DISCHARGED AT *C*, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT *C*

flowing through the pipe section *TB* was found to be 15.6 in. per second. The velocity in the pipe section *AT* was therefore equal to 3.4 in. plus 15.6 in., or 19.0 in. per second.

The velocity heads for the pipe sections *AT* and *TC* were then calculated by the formula

$$h = \frac{v^2}{2g}$$

and found to be 0.015 and 0.466 in., respectively. The difference between these two velocity heads (0.451 in.) was the loss in velocity head while the water was flowing from *R1* to *R3*. Adding the loss of velocity head (0.451) to the loss of pressure head (0.195), the total loss of head was found to be 0.646 in. This loss of head may be divided into three parts: The friction head in the pipe section *R1-T*, the friction head in the tee, and the friction head in the pipe section *T-R3*.

The two pipe friction heads were calculated by the formula

$$h = 0.00685 \frac{v^{1.77}}{d^{1.275}} \text{ (feet of water)}^{4}$$

and found to be 0.177 in. and 0.008 in., respectively. Subtracting these two friction heads from the total loss of head, the friction head in the tee was found to be 0.461 in. Dividing this by the friction head for a 1-in. standard cast-iron elbow (0.014 in.) calculated by the formula

$$h = 0.0141 \frac{v^{1.99}}{d^{0.26}} \text{ (feet of water)}^{4}$$

the friction head in the tee was found to be equivalent to 33 elbows. For this case, 18 per cent of the water flowing in the section *AT* was diverted at right angles through the tee into the section *TC* and the velocity of the water in this section was 3.4 in. per second. This value (33 elbow equivalents) was recorded in Fig. 3.

By varying the percentage of water flowing through the section *TC*, other values were found and recorded in Fig. 3 and a curve drawn through the points. From this curve it appears that when 100 per cent of the water is flowing through the pipe *TC*, the friction head in the tee is equivalent to about 1.8 elbows and that the number of elbow equivalents increases to about 27 when the proportion of water diverted at right angles is decreased from 100 per cent to 20 per cent. It also appears that the number of elbow equivalents approaches infinity as the percentage of water flowing through the section *TC* approaches zero.

The method just described was also used in finding the friction head for the water flowing straight through the tee. The results for this case were recorded in Fig. 4 and the corresponding curve drawn. It appears from this curve that when 100 per cent of the water is flowing straight through the tee the friction head is equivalent to about 0.06 elbow, and that the elbow equivalents are increased to about 14 when the quantity of water flowing straight through the tee is decreased to 20 per cent of that flowing into the tee. In both cases the elbow equivalent friction head is based on the velocity of the water in the pipe through which the water is discharged from the tee.

Check of Results

Since the results found in this investigation differed materially from those which had been expected, it was decided to check them with the apparatus shown in Figs. 5 and 6. In Fig. 5, *T1*, *T2*, and *T3* represent the three 1 x 1 x 1-in. tees under investigation; *A-T1*, a section of pipe 2 ft. long; *T1-T2*, *T2-T3*, *T1-B*, *T2-C*, and *T3-D*, sections of pipe 1 ft. long; *B*, *C*, and *D*, standard 1-in. cast-iron elbows with 1 x 4-in. nipples looking up and reamed to an internal diameter of 1.0625 in.; *R*, a piezometer ring, and *G*, an inclined glass manometer tube with a slope of 1 on 10. The point *O* is the zero point on the manometer tube and is at the same level as the upper ends of the three nipples at *B*, *C*, and *D*.

During a test, the water entered the apparatus at *A* and escaped at *B*, *C*, and *D*, discharging into three separate weighing tanks. The quantities of water escaping from the three nipples were 0.305, 0.473, and 0.518 lb per second, respectively, from which the velocity heads at discharge, the velocities in the six pipe sections, and the velocity head at *R* were calculated. The pressure

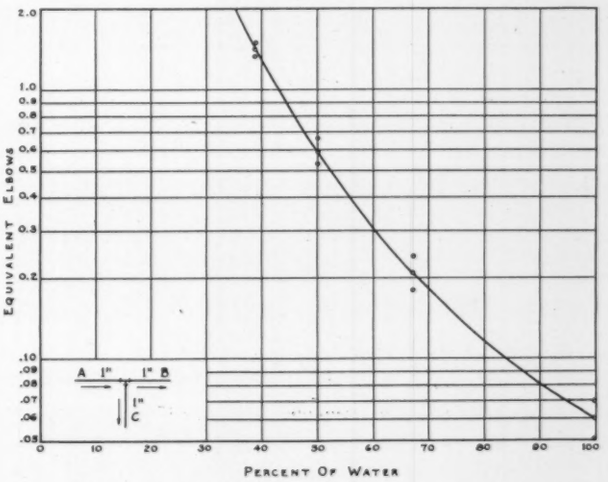


FIG. 4—THE FRICTION HEAD IN A 1x1x1-IN. CAST-IRON TEE WHEN WATER ENTERS AT *A* AND A PORTION IS DISCHARGED AT *B*, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT *B*

[4] The Friction of Water in Pipes and Fittings, by F. E. Giesecke (University of Texas Bulletin No. 1759, October 20, 1917).

head (h) at R was read in the glass tube. Having these data, the friction heads in the three circuits were calculated.

For the data represented in Fig. 5, the friction heads

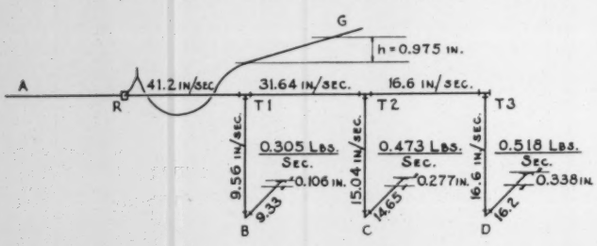


FIG. 5—APPARATUS USED TO CHECK THE FRICTION HEADS IN TEES AS DETERMINED BY THE APPARATUS SHOWN IN FIGS. 1 AND 2

were as follows: Path R-$T1$-B = 3.088 in.; path R-$T2$-C = 2.913 in.; and path R-$T3$-D = 2.857 in. These three values were determined by deducting the sum of the velocity head, pressure head, and elevation head at the discharge from the sum of those heads at R.

The friction head in the path R-$T1$-B includes

Friction in pipe R-$T1$	= 0.700
Friction in pipe $T1$-B	= 0.052
Friction in elbow B	= 0.107
Friction in nipple B	= 0.017
Total	= 0.876

This leaves for the friction in tee $T1$ 3.088 minus 0.876, or 2.212 in.

The friction in tee $T1$ is therefore 20.6 times as large as that in elbow B. In this case 23.3 per cent of the water is diverted at right angles in the tee so the result checks well with the ratio 20 shown by the curve of Fig. 3.

The friction head in the path R-$T2$-C includes

Friction in pipe R-$T1$	= 0.700
Friction in tee $T1$	= 0.220
Friction in pipe $T1$-$T2$	= 0.436
Friction in pipe $T2$-C	= 0.117
Friction in elbow C	= 0.264
Friction in nipple C	= 0.037
Total	= 1.774

This leaves for the friction in tee $T2$ 2.913 minus 1.774, or 1.139 in.

The friction in tee $T2$ is therefore 4.34 times as large as that in elbow C. In this case 48.7 per cent of the water is diverted at right angles in the tee so the result checks well with the ratio 4.76 shown in the curve of Fig. 3.

The friction in the path R-$T3$-D includes

Friction in pipe R-$T1$	= 0.700
Friction in tee $T1$	= 0.220
Friction in pipe $T1$-$T2$	= 0.436
Friction in tee $T2$	= 0.310
Friction in pipe $T2$-$T3$	= 0.137
Friction in pipe $T3$-D	= 0.137
Friction in elbow D	= 0.310
Friction in nipple D	= 0.047
Total	= 2.297

This leaves for the friction in tee $T3$ 2.857 minus 2.297, or 0.560 in. The friction in tee $T3$ is therefore 1.81 times as large as that in elbow D. In this case, 100 per cent of the water is diverted at right angles in the tee so the result checks well with the ratio 1.80 shown in the curve of Fig. 3.

Conclusions

There is a material variation in the friction heads of tees with the percentage of water deflected through the tee, as determined in this investigation and shown by the curves of Figs. 3 and 4. This variation is of importance in the design of piping installations and particularly when applied to hot water heating systems in which slight differences in available heads may produce serious changes in the operation of the heating system.

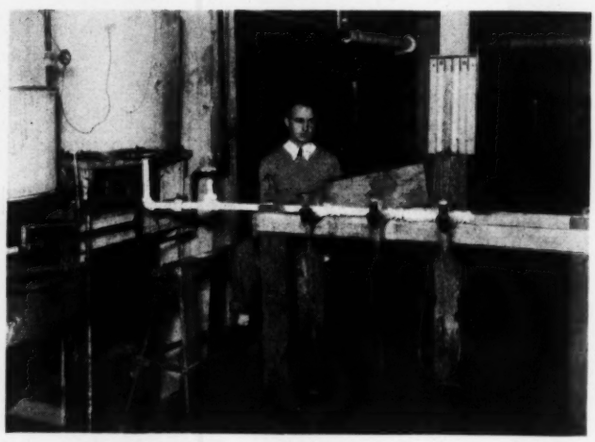


FIG. 6—APPARATUS USED TO CHECK THE FRICTION HEADS IN TEES AS DETERMINED BY THE APPARATUS SHOWN IN FIGS. 1 AND 2

It is important that additional investigations be conducted to determine the friction heads in tees when the water flows in other directions than those discussed in this paper, to determine how the friction head varies with the absolute velocity through the tee, and to determine the difference between the friction heads in a 1 x 1 x 1-in. tee and those in a 1 x 1 x ¾-in. tee.

Investigations along these lines are now under way and results will be reported as early as practicable.

Heat Emission for Other Than Standard Conditions

The A. S. H. V. E. Code for Testing Radiators assumes that the heat emission from cast-iron "radiators" varies with the 1.3 power of the difference between the heating medium within the "radiator" and the air surrounding it. By definition, one square foot of equivalent "radiation" will emit 240 Btu per hour for steam at 215 F and air at 70 F. For other conditions the heat emission will, of course, differ and the lower the temperature of the heating medium within the "radiator," the lower the heat emission. Conversion factors for heat emitted by direct cast-iron heaters, based on the 1.3 power relationship are given on page 206 of THE A. S. H. V. E. GUIDE 1931.

In a one-pipe hot-water heating system, the temperature of the water entering each "radiator" will be lower than that entering the preceding "radiator" in the same circuit. Consequently, as the mean temperature of the water in each "radiator" decreases, the amount of heating surface must be increased, for the same total heat emission. These conversion factors may conveniently be used for this purpose.

Estimating Fuel Consumption

By Paul D. Close[1], New York
MEMBER

MUCH has been written on the subject of fuel consumption, and many formulae have been developed for estimating the seasonal fuel requirements of buildings. Practically all rational methods involve the assumption that the annual heat loss of a building is equal to the calorific value of the fuel consumed when allowance is made for the probable efficiency with which the fuel will be utilized.

The fuel consumption (F) is therefore equal to the total estimated heat loss of the building for the entire heating season (L) divided by the calorific value of the fuel (C) and the efficiency of utilization (E), or

$$F = \frac{L}{CE} \tag{1}$$

The total theoretical heat loss of a building may be calculated with reasonable accuracy from available data such as are given in the A. S. H. V. E. GUIDE. To obtain a reasonably accurate solution of the problem, an allowance should, of course, be made for periods during which no heat is supplied such as at night, and also for the opening of windows and other factors, which would affect the actual fuel consumption.

The total theoretical heat loss (L) is obtained by reducing the calculated hourly heat loss (H) to a one degree basis by dividing by the temperature head ($t - t_o$) upon which the design of the heating system was based, and then multiplying this quantity by the average temperature head ($t - t_a$) during the heating season and the number of hours (N) during the heating season. Therefore—

$$L = \frac{H \times (t - t_a) \times N}{t - t_o} \tag{2}$$

By means of the foregoing principles the following formulae for estimating the theoretical amount of fuel required to heat a building may be derived:

$$T = \frac{H \times (t - t_a) \times N}{(t - t_o) \times C_c \times E_c \times 2000} = \text{tons of coal required} \tag{3}$$

If it is assumed that $N = 5040$, $C_c = 13{,}000$ and $E_c = 0.60$,* then—

$$T = \frac{5040 \times H \times (t - t_a)}{13{,}000 \times 0.60 \times 2000 \times (t - t_o)} = \frac{(t - t_a) \times H}{(t - t_o) \times 3{,}090} \tag{3a}$$

$$P = \frac{H \times (t - t_a) \times N}{(t - t_o) \times C_o \times E_o \times W} = \text{gallons of oil required} \tag{4}$$

If it is assumed that $N = 5040$, $C_c \times W = 141{,}000$ and $E_o = 0.70$*; then—

$$P = \frac{5040 \times (t - t_a) \times H}{141{,}000 \times 0.70 \times (t - t_o)} = \frac{(t - t_a) \times H}{(t - t_o) \times 19.5} \tag{4a}$$

$$G = \frac{H \times (t - t_a) \times N}{(t - t_o) \times C_g \times E_g} = \text{Cubic feet of gas} \tag{5}$$

For *natural gas*, if it is assumed that $C_g = 1{,}000$, $E_g = 0.80$* and $N = 5040$, then—

$$G = \frac{H \times (t - t_a) \times 5040}{(t - t_o) \times 1000 \times 0.80} = \frac{6.31 \times H \times (t - t_a)}{(t - t_o)} \tag{5a}$$

For *manufactured gas*, if $C_g = 535$, then—

$$G = \frac{11.8 \times H \times (t - t_a)}{(t - t_o)} \tag{5b}$$

Infiltration

The hourly heat loss (H) is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i) of the various rooms or spaces to be heated and the total equivalent heating surface ("radiation") required is equal to $\frac{H}{240}$ sq ft. In estimating the fuel consumption of a building of more than one room divided by walls or partitions, it is not correct to use the calculated "radiation" requirements of the building without making the proper allowance for the fact that the heating load at any time does not involve the sum of the infiltration losses of all of the heated spaces of the building but only part of the infiltration losses. This is clearly explained on page 70 of THE A. S. H. V. E. GUIDE 1931, as follows:

> While a heating reserve must be provided to warm inleaking air on the windward side of a building, this does not necessarily mean that the heating plant must be provided with a reserve capacity, since the inleaking air, warmed at once by adequate heating surface in exposed rooms, will move transversely and upwardly through the building, thus relieving other radiators of a part of their load. The actual loss of heat of a building caused by infiltration is not to be confused with the necessity for providing additional heating capacity for a given space. Infiltration is a disturbing factor in the heating of a building, and its maximum effect (maximum in the sense of an average of wind velocity peaks during the heating season, above some reasonably chosen minimum) must be met by a properly distributed reserve of heating capacity, which reserve, however, is not in use at all places at the same time, nor in any one place at all times.

It usually is sufficiently accurate to consider only half of the total infiltration losses of a building having interior walls and partitions, and the value of H in equation (2) would, under these conditions, be equal to $H_t + \frac{H_i}{2}$. If a building has no interior walls or partitions, whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side, and only half of the total crack should be used in computing the

[1] Technical Secretary, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

* See discussion of Heating Efficiencies on page 434.

infiltration for each side and end of the building. Under these conditions it is sufficiently accurate to use the total calculated heat loss (H) for the building.

Certain published formulae for estimating the seasonal fuel consumption of buildings have apparently neglected the aforementioned infiltration factor. In some cases this factor is taken into consideration by means of a constant, but no single constant will accurately allow for all conditions, particularly in view of the fact that in some cases no allowance whatsoever need be made whereas in other cases the allowance may equal 25 per cent of the total heat loss. Of course, it may be argued that the process of estimating the fuel requirements is of such an approximate nature that the infiltration factor can either be neglected or allowed for by means of a single constant in a suitable formula. However, in many cases this item constitutes a large part of the total losses and should therefore be considered. Of course, where the "radiation" is estimated by empirical or rule-of-thumb methods, refinements in approximating fuel consumption are not warranted, but rule-of-thumb methods often lead to unsatisfactory results and should be avoided in heating work where more accurate methods are available.

It should be emphasized that the value of H in equations (3) to (5*b*) inclusive is the total heat loss of the building after making the proper allowance for infiltration.

Symbols

t = inside temperature, degrees Fahrenheit
t_a = average temperature during heating season, degrees Fahrenheit
t_o = outside design temperature, degrees Fahrenheit
C_c = calorific value of coal, Btu per pound
C_g = calorific value of gas, Btu per cubic foot
C_o = calorific value of oil, Btu per pound
D = degree-days per heating season
E_c = efficiency of utilization of coal
E_g = efficiency of utilization of gas
E_o = efficiency of utilization of oil
F = annual fuel consumption
G = cubic feet of gas per heating season
H = calculated heat loss of building based on outside temperature of t_o, Btu per hour
H_i = calculated infiltration losses of building, Btu per hour
H_t = calculated heat transmission losses of building, Btu per hour
L = total heat loss of building for entire heating season, Btu
N = number of hours during heating season corresponding to average temperature t_a
P = gallons of oil per heating season
T = tons of coal per heating season
W = weight of oil per gallon, pounds

Heating Efficiencies

Overall or *house* efficiencies may vary to a considerable extent depending on many factors. The *overall* efficiency will of course differ from the so-called *boiler and grate* efficiency. The following statement on this subject appears on page 44 of THE A. S. H. V. E. GUIDE 1931:

> It should be understood that the heating efficiencies of 60, 70 and 80 per cent * * * are approximate values and apply more specifically where the heating plant is not located in the building heated, and consequently, where there is no regain of the sensible heat of the flue gases, and where the pipe transmission losses may be considerable. Where the heating plant is located within the building to be heated, the *house* or *building* efficiencies for coal, oil and gas may be substantially higher than those specified, although it is even possible that the efficiencies may be lower than these values under varying conditions of operation.

The efficiencies of 60, 70 and 80 per cent quoted in the foregoing excerpt, and which were used for arriving at equations (3*a*), (4*a*), (5*a*) and (5*b*) represent approximate average values obtained from various reliable sources of information on this subject. The values are believed to be reasonably accurate for average conditions, but it should be borne in mind that actual efficiencies in individual cases may differ considerably from these values. Where reliable efficiencies and calorific values for given conditions are available, they may of course be applied by using equations (3), (4) and (5).

Allowance for Non-Heating Periods

Obviously, the theoretical fuel consumption will be reduced considerably by not operating the heating plant at night or during other periods. Allowance for this may be made in either of two ways: (1) by estimating the average inside temperature (t) or (2) by arbitrarily assuming a certain reduction in the fuel consumption.

The first procedure is, of course, the more accurate. If, for example, the daytime temperature is to be 70 F, and the temperature from 12 midnight to 6 A. M. is to be maintained by thermostatic control at 50 F, then the average daily inside temperature (t) will be

$$\frac{18 \times 70 + 6 \times 50}{24} \text{ or } 65 \text{ F.}$$

Strictly speaking, this average inside temperature would only apply when the outside night temperature averages below 50 F, but this fact usually is not of sufficient importance to warrant consideration. If the average outside temperature during the heating season is 30 F, the fuel saving would be approximately

$$100 \times \frac{70 - 65}{70 - 30} = 12.5 \text{ per cent.}$$

In this case, the additional saving in fuel due to the cooling of the air and structural materials to 50 F, would be offset by the heating-up load in the morning.

As to the second procedure, it may be assumed that a saving in the fuel consumption of from 10 to 20 per cent will result if the heat is shut off after working hours, and the building heated to the required temperature during the period of occupancy each day. (See A. S. H. V. E. GUIDE 1931, page 43.) An average allowance of 15 per cent is, no doubt, sufficiently accurate for residences where heat is not supplied during the sleeping hours. For factories where the non-heating periods are longer, the saving may be 20 per cent or more, whereas for hotels 10 per cent would probably be within reason. These, of course, are general statements and wherever possible, the average temperature should be estimated from the proportionate lengths of the occupancy and non-occupancy periods. It should be emphasized that any deviation from the assumed inside temperature will result in a variation in the fuel consumption.

Heat Required to Warm Building

The heat required to warm the cold building and con-

TABLE 1—CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS [a]

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. A	COL. B	COL. C	COL. D	COL. E	COL. F
STATE	CITY	AVERAGE TEMP. OCT. 1ST-MAY 1ST	LOWEST TEMPERATURE	AVERAGE WIND VELOCITY DEC., JAN., FEB., MILES PER HOUR	DIRECTION OF PREVAILING WIND, DEC., JAN., FEB.	STATE	CITY	AVERAGE TEMP. OCT. 1ST MAY 1ST	LOWEST TEMPERATURE	AVERAGE WIND VELOCITY DEC., JAN., FEB., MILES PER HOUR	DIRECTION OF PREVAILING WIND, DEC., JAN., FEB.
Ala.	Mobile	57.7	−1	8.3	N	N. M.	Santa Fe	38.0	−13	7.3	NE
	Birmingham	53.9	−10	8.6	N	N. C.	Raleigh	49.7	−2	7.3	SW
Ariz.	Phoenix	59.5	16	3.9	E		Wilmington	53.1	5	8.9	SW
	Flagstaff	34.9	−25	6.7	SW	N. D.	Bismark	24.5	−45	....	NW
Ark.	Fort Smith	49.5	−15	8.0	E		Devil's Lake	18.9	−44	11.4	W
	Little Rock	51.6	−12	9.9	NW	Ohio	Cleveland	36.9	−17	14.5	SW
Cal.	San Francisco	54.3	29	.....	N		Columbus	39.9	−20	9.3	SW
	Los Angeles	58.6	28	.....	NE	Okla.	Oklahoma City	48.0	−17	12.0	N
Colo.	Denver	39.3	−29	7.4	S	Ore.	Baker	34.1	−20	6.0	SE
	Grand Junction	39.2	−16	5.6	SE		Portland	45.9	−2	6.5	S
Conn.	New Haven	38.0	−14	9.3	N	Pa.	Philadelphia	41.9	−6	11.0	NW
D. C.	Washington	43.2	−15	7.3	NW		Pittsburgh	40.8	−20	13.7	NW
Fla.	Jacksonville	61.9	10	8.2	NE	R. I.	Providence	37.6	−9	14.6	NW
Ga.	Atlanta	51.4	−8	11.8	NW	S. C.	Charleston	56.9	7	11.0	N
	Savannah	58.4	8	8.3	NW		Columbia	53.7	−2	8.0	NE
Idaho	Lewiston	42.5	−13	4.7	E	S. D.	Huron	28.1	−43	11.5	NW
	Pocatello	36.4	−20	9.3	SE		Rapid City	32.3	−34	7.5	W
Ill.	Chicago	36.4	−23	17.0	SW	Tenn.	Knoxville	47.0	−16	6.5	SW
	Springfield	39.9	−24	10.2	NW		Memphis	50.9	−9	9.6	NW
Ind.	Indianapolis	40.2	−25	11.8	S	Texas	El Paso	53.0	−2	10.5	NW
	Evansville	44.1	−15	8.4	S		Fort Worth	54.7	−8	11.0	NW
Iowa	Dubuque	33.9	−32	6.1	NW		San Antonio	60.7	4	8.2	N
	Sioux City	32.1	−35	12.2	NW	Utah	Modena	38.1	−24	8.9	W
Kans.	Concordia	38.9	−25	7.3	N		Salt Lake City	40.0	−20	4.9	SE
	Dodge City	40.2	−26	10.4	NW	Vt.	Burlington	29.3	−27	12.9	S
Ky.	Louisville	45.2	−20	9.3	SW	Va.	Norfolk	49.1	2	9.0	N
La.	New Orleans	61.5	7	9.6	N		Lynchburg	45.2	−7	5.2	NW
	Shreveport	56.2	−5	7.7	SE		Richmond	47.4	−3	7.4	S
Me.	Eastport	31.1	−23	13.8	W	Wash.	Seattle	45.3	3	9.1	SE
	Portland	33.6	−17	10.1	NW		Spokane	37.5	−30	....	SW
Md.	Baltimore	43.6	−7	7.2	NW	W. Va.	Elkins	38.8	−21	4.8	W
Mass.	Boston	37.6	−13	11.7	W		Parkersburg	41.9	−27	6.6	S
Mich.	Alpena	29.1	−27	11.3	W	Wis.	Green Bay	28.6	−36	12.8	SW
	Detroit	35.4	−24	13.1	SW		La Crosse	31.2	−43	5.6	NW
	Marquette	27.6	−27	11.4	NW		Milwaukee	33.0	−25	11.7	W
Minn.	Duluth	25.1	−41	11.1	SW	Wyo.	Sheridan	31.0	−45	5.3	NW
	Minneapolis	29.6	−33	11.5	NW		Lander	28.9	−36	3.0	NE
Miss.	Vicksburg	56.0	−1	7.6	SE	Alta.	Edmonton	23.3	−57	4.5	W
Mo.	St. Joseph	40.3	−24	9.1	NW	B. C.	Victoria	43.8	−2	8.9	N
	St. Louis	43.3	−22	11.8	NW		Vancouver	41.7	2	4.2	E
	Springfield	43.0	−29	11.3	SE	Man.	Winnipeg	17.2	−46	12.4	SW
Mont.	Billings	34.7	−49	....	W	N. B.	Fredericton	27.1	−35	8.7	NW
	Havre	27.7	−57	8.7	SW	N. S.	Yarmouth	35.5	−12	13.0	NW
Nebr.	Lincoln	37.0	−29	10.9	N	Ont.	London	32.5	−26	....	....
	North Platte	34.6	−35	9.0	W		Ottawa	26.9	−33	7.5	W
Nev.	Tonopah	39.6	−7	9.9	SE		Pt. Arthur	21.6	−51	....	....
	Winnemucca	37.9	−28	9.5	NE		Toronto	32.0	−26	13.5	SW
N. H.	Concord	33.4	−35	6.0	NW	P. E. I.	Charlottetown	30.1	−23	8.7	NW
N. J.	Atlantic City	41.6	−7	10.6	NW	Que.	Montreal	27.4	−27	15.4	SW
N. Y.	Albany	35.1	−24	7.9	S		Quebec	24.4	−34	15.0	SW
	Buffalo	34.7	−14	17.7	W	Sask.	Prince Albert	14.7	−70	3.2	SW
	New York	40.3	−6	13.3	NW	Yukon.	Dawson	1.6	−68	....	....

[a] Reprinted from A. S. H. V. E. GUIDE 1931.

tents is a factor to be considered. Under certain conditions, the cooling of the structure and contents will, to some extent, compensate for the heat required to rewarm the building. For example, if the building is under thermostatic control and the day and night temperatures are say 70 F and 50 F respectively, there will be a period during which no heat will be called for while the building is cooling to 50 F and the saving resulting therefrom will correspond to the additional heat required to bring the building and contents back to the daytime temperature. If in estimating the fuel consumption the average daily inside temperature is based on the proper

day and night temperatures and periods, the heat required to warm the structure may be neglected.

Where irregular conditions are involved it may be desirable to actually calculate the heat required to warm the building structure and contents for the number of times during the heating season the heating plant would not be in operation and to add this quantity to the fuel required for the number of hours during which the building is heated. The greater the heat capacity of the structure the greater will be the relative importance of this item. For structures of low heat capacity, such as frame buildings, this factor usually may be neglected.

Other Factors

There are many factors which would be likely to affect the theoretical fuel requirements of a building such as the opening of windows, abnormal inside temperatures, other heat sources, etc. In many cases it is difficult accurately to evaluate these factors, particularly in the case of open windows, and the results are correspondingly less accurate. The degree of refinement of the calculations should, of course, be consistent with the conditions involved. If the heat loss from the boiler and piping does not warm the building or is not included in H, the proper allowance should be made. In selecting a boiler, this allowance is frequently assumed to be 25 per cent of the total heat loss of the building, but in estimating fuel requirements, the more accurate procedure of computing the pipe and boiler losses should be used, unless this item is likely to be outweighed by other less tangible factors.

Where temperature control is installed the fuel consumption can obviously be predetermined with greater accuracy than where no such control has been provided. In fact the calculated requirements agree to a remarkable degree in many cases with the actual consumption. This has been particularly true of gas-fired installations, with which effective temperature regulation usually is possible.

Where other heat sources are available it quite often is possible accurately to allow for the reduction in the fuel consumption resulting therefrom. These sources include the heat supplied by persons, lights, motors and machinery, and should also be ascertained in the case of theaters, assembly halls and industrial plants. In many cases these heat sources should not be allowed to affect the size of the installation of heating equipment, although they may have a marked effect upon the fuel consumption. In residences this factor usually may be entirely neglected.

Example 1. A small factory building located in Philadelphia is to be heated to 60 F between the hours of 7 a. m. and 7 p. m., and to 50 F during the remaining hours. The calculated hourly heat loss, based on a design temperature of —6 F, is 500,000 Btu. If coal having a calorific value of 12,500 Btu per pound is fired, and the over-all heating efficiency is assumed to be 60 per cent, how many tons of coal will be required for a normal heating season, neglecting other heat sources, and any loss of heat through open windows?

Solution. Since there are no partitions in this building, the entire heat loss is considered. The average outside temperature during the heating season (t_a) is 41.9 F (see Table 1); $t = 60$ F; $N = 5040$; $H = 500{,}000$; $t - t_o = 66$ F; $C_e = 12{,}500$; $E_e = 0.60$. Substituting the values in equation (3):

TABLE 2—DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA [a]

Col. A	Col. B	Col. C	Col. A	Col. B	Col. C
State or Province	City	Degree-Days	State or Province	City	Degree-Days
Ala.	Mobile	1,439	Nev.	Tonapah	6,069
	Birmingham	2,527		Winnemucca	6,266
Ariz.	Phoenix	1,446	N. H.	Concord	7,335
	Flagstaff	10,913	N. J.	Atlantic City	5,250
Ark.	Hot Springs	2,665	N. Y.	Albany	6,542
	Little Rock	2,861		Buffalo	6,750
Cal.	San Francisco	3,450		New York	5,348
	Los Angeles	1,517	N. M.	Santa Fe	6,064
Colo.	Denver	5,880	N. C.	Raleigh	3,287
	Grand Junction	5,570		Wilmington	2,493
Conn.	New Haven	6,039	N. D.	Bismark	8,498
D. C.	Washington	4,562		Grand Forks	9,724
Fla.	Jacksonville	1,080	Ohio	Cleveland	6,096
Ga.	Atlanta	2,880		Columbus	5,426
	Savannah	1,517	Okla.	Oklahoma City	3,827
Idaho	Lewiston	4,924	Ore.	Portland	4,449
	Pocatello	6,459		Salem	4,629
Ill.	Chicago	6,300	Pa.	Philadelphia	4,950
	Springfield	5,495		Pittsburgh	5,327
Ind.	Indianapolis	5,331	R. I.	Providence	6,111
	Evansville	3,355	S. C.	Charleston	1,770
Iowa	Dubuque	6,744		Columbus	2,600
	Des Moines	6,464	S. D.	Pierre	7,213
Kans.	Topeka	5,282		Sioux Falls	7,683
	Dodge City	5,035	Tenn.	Knoxville	3,517
Ky.	Louisville	4,366		Nashville	3,550
La.	New Orleans	1,044	Texas	El Paso	1,912
	Shreveport	2,097		Galveston	1,050
Me.	Eastport	8,676		San Antonio	1,362
	Portland	7,267	Utah	Salt Lake City	5,358
Md.	Baltimore	4,591		Logan	6,750
Mass.	Boston	6,055	Vt.	Burlington	8,123
Mich.	Alpena	8,319	Va.	Richmond	3,789
	Detroit	6,202		Lynchburg	3,849
	Marquette	8,866		Danville	3,316
Minn.	Duluth	9,650		Norfolk	3,349
	Minneapolis	7,953	Wash.	Seattle	5,156
Miss.	Jackson	1,920		Spokane	6,085
Mo.	St. Joseph	5,289	W. Va.	Elkins	5,813
	St. Louis	4,583		Parkersburg	4,884
	Springfield	4,650	Wis.	Green Bay	8,201
Mont.	Billings	6,983		La Crosse	7,309
	Havre	8,608		Milwaukee	7,366
Nebr.	Lincoln	6,231	Wyo.	Sheridan	8,113
	North Platte	6,479		Cheyenne	7,360
B. C.	Victoria	5,777	Ont.	Toronto	7,732
	Vancouver	5,976	Que.	Montreal	8,705
	Kamloops	6,724		Quebec	8,628
Alta.	Medicine Hat	8,152	N. B.	Fredericton	9,099
Sask.	Qu' Appelle	11,261	N. S.	Yarmouth	7,694
Man.	Winnipeg	11,166	P. E. I.	Charlottetown	8,485
Ont.	Port Arthur	10,803			

[a] Reprinted from A. S. H. V. E. GUIDE 1931.

$$T = \frac{500{,}000 \times 18.1 \times 5040}{66 \times 12{,}500 \times 0.60 \times 2000} = 46 \text{ tons}$$

Inasmuch as the building will be heated to 50 F at night, the average inside temperature (at the breathing line) will be 55 F, and the percentage saving will be $\frac{60 - 55}{60 - 41.9} = 0.276$ or 27.6 pe

cent. The net fuel consumption will therefore be $46 - 0.276 \times 46$ or 33.3 tons.

Degree-Day Method

A very useful unit for estimating fuel consumption—particularly for residences—is the well-known degree-day which is defined as a day on which the mean temperature is one degree below 65 F, or 64 F. Degree-days for various cities in the United States and Canada are given in Table 2. The term degree-day originated in the gas industry and was later standardized by the *American Gas Association.*

The base of 65 F is used for an inside temperature of 70 F. This base was chosen because it was demonstrated, by means of data collected from numerous installations, that 65 F usually is the maximum temperature at which fuel is burned to supply heat for a residence. It was also found that the fuel consumed varied almost directly with the difference between 65 F and the outside temperature. In other words, when the temperature is 20 deg below 65 F, approximately twice as much fuel will be consumed as on a day when the temperature is 10 deg below 65 F for a specified inside temperature of 70 F. This is probably due to the fact that the average inside temperature for the 24-hour period is about 65 F for a 70-F daytime temperature.

If the inside temperature were maintained at 70 F throughout the 24 hours of the day, then the base of 65 F would probably be in error. It must be borne in mind, however, that while the heat head is the difference between the inside temperature of say 70 F, and the outside temperature, a lower temperature than 70 F will usually be maintained at night and the base of 65 F will therefore allow for this condition. As already indicated, a temperature of 50 F from midnight to 6 a.m. will reduce the 24-hour average from 70 to 65 F. It is important to note that the degree-day applies specifically to an inside temperature of 70 F, which is usually considered the standard temperature for residences, and it should also be noted that allowance is automatically made for the lower night-time temperature, although this allowance is constant for any given locality.

In equations (3), (4) and (5), the quantity $(t - t_a) \times N$ is equivalent to the number of degree-days in a heating season multiplied by 24, when the average daily value of t is 65 F. Therefore

$$(t - t_a) \times N = 24\ D. \qquad (6)$$

In the degree-day system the heating season begins when the normal outside temperature becomes 65 F and ends when the normal temperature again becomes 65 F in the spring. The number of hours in the heating season (N) in equations (3*a*), (4*a*), (5*a*) and (5*b*) was assumed to be constant and to have a duration of 210 days or 5040 hours, and when using these equations the average temperature should correspond with this period of time, or approximately from October 1 to May 1, which is the period for which heating season temperature averages are usually given. In this respect the degree-day usually is a more accurate unit than the quantity $(t - t_a) \times N$ because the heating season, which begins and ends with a temperature of 65 F, may actually be of longer or shorter duration than 210 days. However, in the northern half of the United States, the duration of the heating season varies little from the 210-day period, and if the average temperature corresponding to this period is used, the result will be approximately the same as it would be if the actual heating season were used together with the average outside temperature for the actual heating season. In other words, the product of $(t - t_a)$ and N will be practically the same in either case. This discussion is given merely to point out that the selection of a heating season (in the northern states) of 210 days is approximately correct even though the actual heating season may be slightly more or less than 210 days.

Substituting the value of $(t - t_a) \times N$ from equation (6) in equations (3), (4) and (5), the following formulae are obtained:

$$P = \frac{H \times D}{(t - t_o) \times C_c \times E_c \times 83.3} = \text{tons of coal required} \qquad (7)$$

If $C_c = 13{,}000$ and $E_c = 0.60$,* then

$$T = \frac{H \times D}{(t - t_o) \times 650{,}000} \qquad (7a)$$

$$P = \frac{H \times D \times 24}{(t - t_o) \times C_o \times E_o \times W} = \text{gallons of oil required} \qquad (8)$$

If $C_o \times W = 141{,}000$ and $E_o = 0.70$,* then

$$P = \frac{H \times D}{(t - t_o) \times 2430} \qquad (8a)$$

$$G = \frac{H \times D \times 24}{(t - t_o) \times C_g \times E_g} = \text{cubic feet of gas} \qquad (9)$$

For *natural gas,* if $C_g = 1000$ and $E_g = 0.80$,* then

$$G = \frac{0.03 \times H \times D}{(t - t_o)} \qquad (9a)$$

For *manufactured gas,* if $C_g = 535$, and $E_g = 0.80$,* then

$$G = \frac{0.056 \times H \times D}{(t - t_o)} \qquad (9b)$$

Example 2. The calculated hourly heat loss of a residence located in Chicago is 127,000 Btu, which includes 28,000 Btu for infiltration. The design temperatures are —8 F and 70 F. The normal heating season is assumed to be 210 days (5,040 hours) and the average temperature during this period is 36.4 F. (See Table 1). The building is to be heated with oil fuel having a calorific value of 141,000 Btu per gallon. The heating efficiency is assumed to be 70 per cent. Thermostatic control is to be used and a temperature of 55 F is to be maintained from 11 p.m. to 7 a.m. How many gallons of oil will be required during a normal heating season if the loss of heat through open windows is neglected?

Solution. The maximum hourly heat loss will be $127{,}000 - \frac{28{,}000}{2} = 113{,}000$ Btu $= H$. Substituting the proper values in equation (4):

$$P = \frac{113{,}000 \times (70 - 36.4) \times 5040}{141{,}000 \times 0.70 \times [70 - (-8)]} = 2490 \text{ gal}$$

The average inside temperature will be

$$\frac{70 \times 16 + 55 \times 8}{24} = 65 \text{ F},$$

and the fuel saving due to this fact will be

* See discussion of Heating Efficiencies on page 434.

$$\frac{70-65}{70-36.4} = 0.149 \text{ or } 14.9 \text{ per cent.}$$

Hence, the net fuel consumption will be $2490 - 0.149 \times 2490 = 2120$ gal.

The normal number of degree-days for Chicago is 6300. Substituting in equation (8) and solving by the degree-day method:

$$P = \frac{113{,}000 \times 6300 \times 24}{78 \times 141{,}000 \times 0.70} = 2220 \text{ gal.}$$

No allowance need be made for the average temperature of 65 F since this is taken care of by the selection of a base of 65 F for the degree-day, as already explained. It will be noted that the two methods check within 5 per cent in this case. If the average daily inside temperature in the first solution had been 66.4 F instead of 65 F, the two methods would have checked exactly.

Industrial Degree-Day

Since the standard degree-day is intended for an inside temperature of 70 F, it is particularly convenient for solving residence problems. Where the temperature differs greatly from 70 F, the standard degree-day cannot be accurately applied. Consequently, the industrial degree-day[2] has been developed and values have been derived for two bases, namely 55 F and 45 F, intended for inside temperatures of 60 F and 50 F, respectively.

It is obvious that the degree-days for any base other than 65 F cannot be determined by taking a direct proportion. For example, there are 5348 degree-days for New York for the 65 F base, whereas there are only 3089 degree-days for the 55 F base, which is considerably less than $\frac{55}{65}$ of 5348.

There are $\frac{3089}{5348}$ or 57.8 per cent as many degree-days for New York for the 55 F base, and $\frac{1098}{5348}$ or 20.4 per cent as many degree-days for the 45 F base. These are in the proportion of 100 to 57.8 to 20.4. The point to be emphasized is that there is a considerable spread between these three bases, and consequently there would be an appreciable error if the actual basis to be used in a certain case would be midway between any two of the three bases for which degree-day values are at present available. Since the correction cannot be made on a proportionate basis, it would be more accurate in the majority of cases involving inside temperatures other than 70, 60 or 50, to apply the method submitted in the first part of the article, using equations (3), (4) and (5). The author recommends the use of this method for specified inside temperatures other than 70, 60 or 50, until such time as complete degree-days are available for the entire range of temperatures used in heating work.

[2] See Heating and Ventilating Degree-Day Handbook.

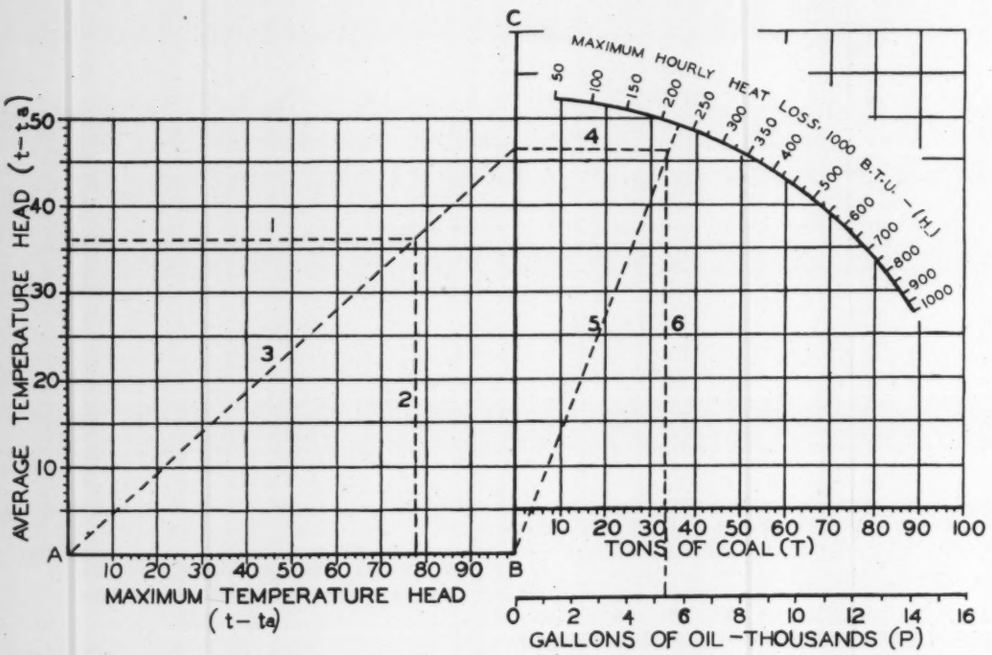


FIG. 1—CHART FOR ESTIMATING FUEL CONSUMPTION IN TONS OF COAL AND GALLONS OF OIL

Rough Approximations

There are many short cuts in use for quickly approximating the fuel consumption of a building where a more accurate analysis is not necessary. These methods include the use of constants involving the fuel consumption per square foot of grate area, per square foot of radiation installed, or per 1,000 cu ft of building to be heated.

Fig. 1 is a convenient chart for quickly estimating the normal fuel consumption in tons of coal or gallons of oil. This chart is based on equations (3*a*) and (4*a*), involving calorific values of 13,000 Btu per pound and 141,000 Btu per gallon for coal and oil respectively, and fixed efficiencies of 60 per cent and 70 per cent respectively.

A problem may be solved merely by drawing lines in the order indicated by the numbers adjoining the dotted lines. The maximum hourly heat loss should, of course, be that upon which the design of the heating

system was based, minus the proper allowance for infiltration, as previously stated. If the heat loss is not known, but the amount of heating surface to be installed is available, the maximum hourly heat loss can, of course, be readily obtained by multiplying the number of square feet of heating surface by 240 for steam, and 150 for hot water,[3] provided the "radiation" has been accurately determined. If the "radiation" has been accurately determined, the infiltration factor may be allowed for by deducting about 10 per cent from the estimated fuel requirements, except in the case of buildings without interior partitions, in which case no allowance for infiltration is necessary. If desirable, an allowance may also be made for the maintenance of lower temperatures at night than in the day time by first determining the average daily temperature, and then estimating the proportionate fuel saving resulting therefrom.

The *American Gas Association* has made a very extensive study of the subject of fuel consumption in connection with gas-heated residences. It has been possible to make a very accurate analysis of the fuel required because of the fact that meter readings of the gas consumption were obtainable for a large number of buildings of different types.

Fig. 2 is taken from the 3rd edition of Industrial Gas Series on House Heating, published by the *American Gas Association*, and indicates the average gas consumption per degree-day for various heat contents. While the fuel consumption in individual cases may vary somewhat from the curve values, these average values are sufficiently accurate for estimating purposes and give very satisfactory results.

The value generally used in the manufactured gas industry for residences is 0.21 cu ft per degree-day per square foot of equivalent steam radiation (240 Btu) based on the theoretical requirements. A correction for warmer climates is necessary and it is customary to gradually increase the fuel consumption below 3,000 degree-days to about 20 per cent more at 1,000 degree-days.

For hot water or warm air the fuel consumption is about 0.19 cu ft per degree-day per square foot of equivalent *steam* radiation, that is, per 240 Btu per hour. The actual requirements likewise increase with hot water or warm air systems as the number of degree-days decreases below 3,000. For larger installations, that is, 1,000 sq ft of theoretical "radiation" and above, there is an increase in efficiency, and a consequent decrease in the fuel consumption per degree-day per square foot of heating surface.

It is interesting to note that the results obtained by the *American Gas Association* agree satisfactorily with the results obtained by means of equation (9). For example, if the maximum temperature head $(t - t_o)$ is 70 deg, the calorific value of the fuel, 535 Btu per cubic foot, and the heating efficiency 77 per cent, the fuel consumption will be 0.20 cu ft of gas per degree-day per square foot of equivalent steam radiation. The average efficiency on a large number of gas-fired residence installations was 77 per cent, and, as previously stated, the efficiency increases to a certain extent with an increase in the size of the installation. Using an efficiency of 80 per cent for larger installations, the fuel consumption would be 0.192 cu ft of gas per degree-day per square foot of installed radiation for manufactured gas, according to equation (9). The fuel consumption per square foot of "radiation" would be slightly higher if the amount of "radiation" used in the calculations represented that required to take care of only half the infiltration losses plus the transmission losses.

Steam consumption factors for various types of buildings located in New York City are given on page 378 of THE A. S. H. V. E. GUIDE 1931. Considerable additional data on the heating requirements of various types of buildings in a number of cities may be found in the section on Utilization in the Handbook of the *National District Heating Association*. Excellent fuel consumption formulae for coal have also been developed by E. F. Tweedy and D. S. Boyden. See Cost of Operating Heating Plants, by G. W. Martin (A. S. H. V. E. TRANSACTIONS, Vol. 22, 1916).

[3] The heat emission varies with the air and water temperatures. Conversion factors are given on page 206 of THE GUIDE 1931. For a 70-F room temperature and a water temperature of 170 F, the factor is 0.617, which means that the heat emission under these conditions is 61.7 per cent of that under standard conditions, or 0.617 x 240 = 148 Btu per square foot.

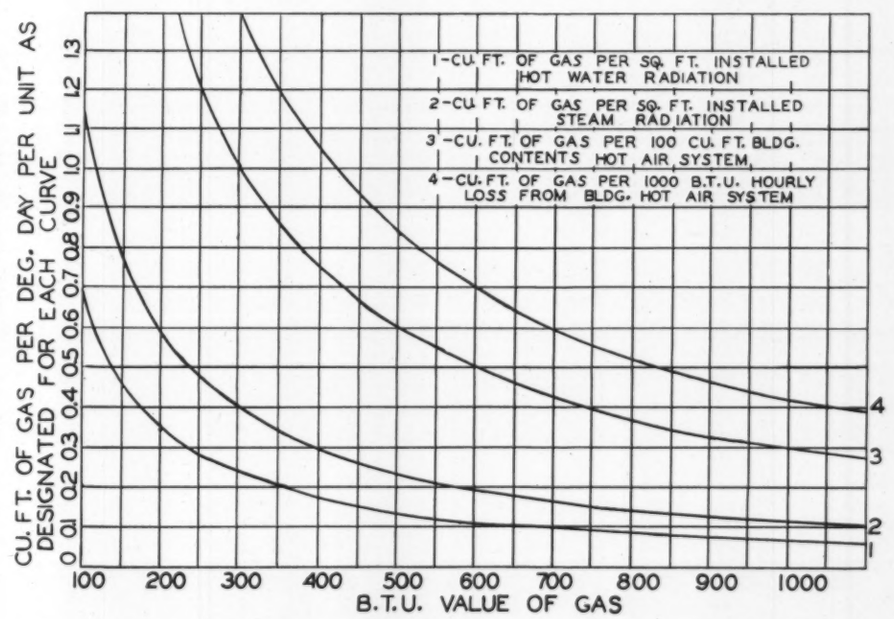
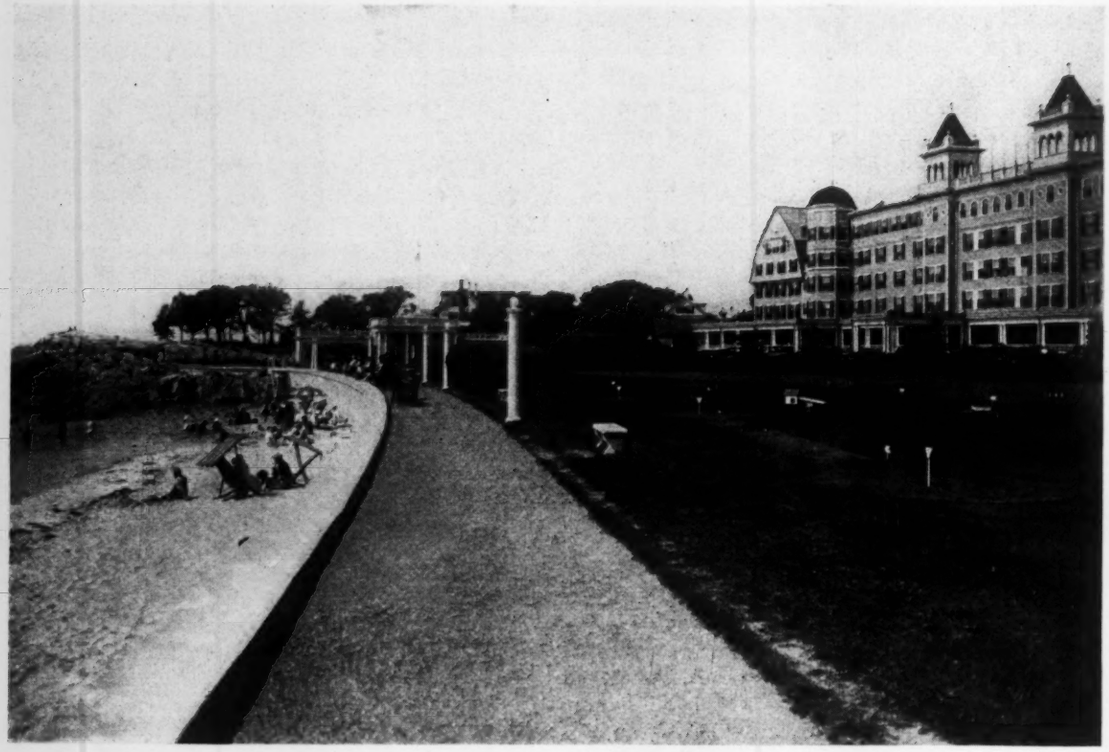

1-CU. FT. OF GAS PER SQ. FT. INSTALLED HOT WATER RADIATION
2-CU. FT. OF GAS PER SQ. FT. INSTALLED STEAM RADIATION
3-CU. FT. OF GAS PER 100 CU. FT. BLDG. CONTENTS HOT AIR SYSTEM
4-CU. FT. OF GAS PER 1000 B.T.U. HOURLY LOSS FROM BLDG. HOT AIR SYSTEM
CU. FT. OF GAS PER DEG. DAY PER UNIT AS DESIGNATED FOR EACH CURVE
0 01 02 03 04 05 06 07 08 09 10 11 12 13
100 200 300 400 500 600 700 800 900 1000
B.T.U. VALUE OF GAS
1
2
3
4

FIG. 2—CHART GIVING GAS REQUIREMENTS PER DEGREE-DAY FOR VARIOUS CALORIFIC VALUES OF FUEL AND FOR DIFFERENT HEATING SYSTEMS[a]

[a] This chart is based on an inside temperature of 70 F and an outside temperature of zero. If the radiation is installed on the basis of any other temperature difference, multiply the result obtained from this chart by 70, and divide by the actual temperature difference.



BEACH AT NEW OCEAN HOUSE

Massachusetts Members Await A.S.H.V.E. at Swampscott

June 22 to 25

SWAMPSCOTT offers four days of enjoyment June 22 to 25, when the Semi-Annual Meeting of the Society will be held at the New Ocean House.

The first day, June 22, has been set aside for registration and various committee meetings. Important among these are the Council meeting at 2 p. m. and the meeting of the Committee on Research at 3 p. m.

The first of the entertainment events commences at 8 p. m., Monday, June 22, when there will be a concert in the hotel lounge followed by an informal reception and dance from 9 to 11 p. m.

The technical sessions will begin at 9:30 a. m. Tuesday, June 23, at which various papers and reports will be presented during the three hour morning session. In accordance with the usual custom, each afternoon, June 23 to 25, will be reserved for a program of sports and entertainment.

The details of the various events will come to members by mail and through announcements in THE JOURNAL, and will include complete information regarding hotel accommodations, automobile routes, train schedules and information about historic New England.

The Massachusetts Chapter has sent each Society member a cordial invitation to attend the Semi-Annual Meeting 1931, which is to be held at the New Ocean House located at Swampscott, 12 miles north of Boston, at the gateway to the famous North Shore Drive. The hotel, which consists of the main house and the new part connecting directly with it, known as Puritan Hall, is situated directly on the ocean. It fronts upon a broad, curving, sandy beach, and commands a beautiful view of Massachusetts Bay and the Atlantic Ocean.

The proximity of Swampscott to so many centers of historical and educational significance, such as Salem, Marblehead, Lexington, Concord, Plymouth, Cambridge and Boston, contributes in making this location attractive as a meeting place. Every recreational feature is available, including golf, tennis, bathing, riding, motoring, boating, fishing and motion pictures. There is a modified golf course on the grounds of the hotel for the use of guests. This course of 1,000 yards is built with regulation greens and hazards. In addition, privileges have been obtained at several private courses, all located within short motoring distance of the hotel.

The arrangements for the meeting are in charge of the following committee:

D. S. Boyden, *General Chairman*
W. T. Jones, *Vice Chairman*
J. S. Webb, *Chairman,*
Reception and Registration Committee
Alfred Kellogg, *Chairman,*
Hotel and Meetings Committee
W. T. Jones, *Chairman,*
Finance Committee
Mrs. E. A. Dussosoit, *Chairman*
Ladies' Entertainment Committee

E. A. Dussosoit, *Chairman,*
Sports Committee
Leslie Clough, *Chairman,*
Publicity Committee
J. W. Brinton, *Chairman,*
Entertainment Committee
J. F. Tuttle, *Chairman,*
Golf Committee
T. F. McCoy, *Chairman,*
Transportation Committee

The headquarters for the meeting, New Ocean House, is operated under the American plan, and rates include room and board. A variety of rooms are available as follows:

HOTEL RATES PER PERSON PER DAY

Single rooms with bath	$9.00 and $10.00
Single rooms with running water	8.00 and 9.00
Double rooms with bath, twin beds	8.00 and 9.00
Double rooms with bath, three beds	7.00
Double rooms with running water, twin beds	7.00

(Garage space in concrete fireproof garage, $1.00 per day.)

Swampscott is reached by excellent roads from all parts of the United States, and those who drive and desire to see points of interest in Boston and vicinity will find a convenient information booth maintained by the Massachusetts Chapter members at the Hotel Statler in Boston, June 21 and 22.

The train service to Swampscott comes direct from the West over the Boston and Maine R. R. and from the South via North Station, Boston.

Complete details regarding roads, train schedules and reduced railroad rates will be sent to each member well in advance of the meeting.

This Semi-Annual Meeting of the Society can be made a real vacation meeting as Swampscott and vicinity abounds in historic places, many of which have already celebrated the 300th anniversary of their founding.

In addition to the many attractive places on Massachusetts North Shore, such as Plymouth, Gloucester, and Marblehead, there is an excellent opportunity for members and their families to motor through Boston and see Old Ironsides at the Navy Yard, Paul Revere House, Old State House, the Birthplace of Benjamin Franklin, John Hancock's House, Fanueil Hall, Bunker Hill Monument and hundreds of other historic places associated with the early history of the United States.

A drive along the North Shore in the direction of Cape Ann brings the visitor to Salem, the birthplace of Nathaniel Hawthorne, the House of Seven Gables, built in 1769, and the Witch House, built in 1675. In Marblehead will be found the world famous painting, "The Spirit of '76," which hangs in Abbott Hall. Here also is the old Powder House, built in 1755, which was used in the Revolutionary War and the War of 1812. Marblehead is also a yachting center and its harbor is always dotted with a great fleet of sailing vessels.

Gloucester is a celebrated fishing port and today maintains its original character, though many other nearby towns have become industrial centers and summer resorts.

Nearby points of interest are Lexington, Concord and Sudbury, and these scenes of American history are the mecca of thousands. At Lexington the Minute-Man Monument may be found and a boulder marks the line of the minute-men. At Concord Bridge was fired the shot "heard around the world," and the Minute-Man Statue commemorates the Battle of Concord.

Wayside Inn at Sudbury, built in 1686, is famed because of Henry Wadsworth Longfellow's Tales of a Wayside Inn, and here are preserved many authentic relics of our forefathers' time.

South of Swampscott there is a large expanse of coast line at the tip of which is Cape Cod. Plymouth and Provincetown offer unusual attractions to the historically inclined visitor.

Because New England is a great center of educational institutions, members who are interested in the Society's research program will find cooperative work being done at Harvard University School of Medicine, in Boston, and in the industrial sections of Boston there are many manufacturing plants producing equipment used by the heating and ventilating profession.

A trip to New England for the Semi-Annual Meeting at Swampscott should be put on the program of ***things to be done this summer*** by the A. S. H. V. E. members.

TECHNICAL PROGRAM

SEMI-ANNUAL MEETING, 1931

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

New Ocean House, Swampscott, Mass., June 22-25, 1931

Monday, June 22, 1931

9:30 a. m.	Registration.
12:30 p. m.	Luncheon (Officers, Council and Authors).
2:00 p. m.	Council Meeting.
3:00 p. m.	Meeting Committee on Research.
3:00 p. m.	Meeting Nominating Committee.
4:00 p. m.	Meeting Guide Publication Committee.

GOLF COURSE AT NEW OCEAN HOUSE

Tuesday, June 23, 1931

8:30 a. m. Registration.

9:30 a. m. Technical Session.

Greeting by D. S. Boyden, Chairman Committee on Arrangements.

Response by Pres. W. H. Carrier.

Report of Committee on Increase of Membership, by C. W. Farrar, Chairman.

Flow of Steam through Orifices into Radiators, by S. S. Sanford and C. B. Sprenger.

Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett.

Pipe and Orifice Sizes for Small Gravity Circulation Hot Water Heating Systems, by E. G. Smith.

Report of Guide Publication Committee, by D. S. Boyden, Chairman.

Report of Committee on Testing and Rating Unit Ventilators, by John Howatt, Chairman.

Wednesday, June 24, 1931

9:30 a. m. Technical Session.

Report of Committee on Research, by C. V. Haynes, Chairman.

Reports of Technical Advisory Committees.

Infiltration through Double Hung Wood Windows, by G. L. Larson.

Heat and Moisture Losses from the Bodies of Men Working and Their Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant.

The Measurement of the Flow of Air through Registers and Grilles, by L. E. Davies.

Report of Committee on Ventilation Standards, by W. H. Driscoll, Chairman.

Thursday, June 25, 1931

9:30 a. m. Technical Session.

Report of Committee on Rating and Testing Concealed Gravity Type Radiation, by R. N. Trane, Chairman.

Performance Characteristics of Cast-Iron and Non-Ferrous Radiators, by A. C. Willard.

Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar.

Heat Transmission Through Wall Structures, by F. B. Rowley.

Local Chapter Reports

Kansas City

April 14, 1931. A meeting of the Kansas City Chapter was held at noon at the Kansas City Athletic Club and was attended by 40 members. W. H. Carrier, National President of the Society, was the guest of honor. Those who could not attend the evening meeting at Lawrence, Kans., had the opportunity of meeting Mr. Carrier at this luncheon.

Late in the afternoon about 100 members of the Society, together with architects and engineers of other Societies, traveled to Lawrence, Kans., and were the guests of the School of Engineering and Architecture at Kansas University. Dinner was served to over 200, and Mr. Carrier's talk at Marvin Hall Auditorium on the Campus was given to a considerably greater number.

The subject of his interesting discourse was Servicing the Human Power Plant, and a very clear conception was given of the constructive work being carried on at the A. S. H. V. E. Research Laboratory, as well as at the institutions with which the Society maintains cooperative agreements. A detailed explanation of the physiological processes of the human body was presented by Mr. Carrier, as well as the relation of temperature, humidity and air motion to comfort and health.

His talk was illustrated by slides which graphically depicted the fundamental relationships involved. His explanation of the metabolic processes of the human body was extremely interesting. Mr. Carrier also pointed out the importance of the proper distribution of conditioned air, and described some of the difficulties encountered in endeavoring to obtain satisfactory distribution without objectionable drafts.

After Mr. Carrier's talk, many questions were asked, which were readily answered by him.

Illinois

April 13, 1931. One of the largest gatherings in the history of the Illinois Chapter assembled in the Bal Tabarin, Hotel Sherman, Chicago, where 162 members and guests had the pleasure of hearing Pres. W. H. Carrier render an address on Servicing the Human Power Plant.

A. G. Sutcliffe presided and various business matters of the Chapter were acted upon promptly including the election of ten applicants for membership as follows: S. I. Rottmayer, P. F. Callaghan, W. G. Hillen, H. B. Johns, A. H. Goelz, J. J. Hagerman, J. H. Bailey, A. M. Smail, G. H. Meffert and R. P. Rasmussen.

President Sutcliffe then introduced the 14 National officers and members of the Council who were present, the six Past Presidents of the Society and the members of the Committee on Research and the Committee on Ventilation Standards.

It was suggested by Messrs. Hale, Heckel and Thomas that Article 7 of the Chapter Constitution relating to the election of officers be amended and ballots were distributed to the members, thus placing the proposal before the meeting for vote. C. W. DeLand gave a resume of a letter from H. R. Linn after which it was decided by a rising vote to defer action on the proposed amendment until a special business meeting in May.

R. E. Moore, W. B. Graves and L. J. Pitcher were appointed by President Sutcliffe as a Committee to arrange for the Annual Golf Tournament in May.

The introduction of President Carrier was made by John Howatt, in a very effective manner.

President Carrier discussed the progress in the general art of heating and ventilating from which he said has sprung the higher specialized art of air conditioning. His talk was confined to air conditioning as it affects human health in the home, rather than its application to industrial arts or manufacturing processes.

He compared the human body and its requirements with a power plant, pointing out that as in a power plant it is necessary not only to supply oxygen to the body, and to remove the products of combustion, carbon dioxide and water vapor, it is also necessary to remove a large proportion of the waste heat present in every thermodynamic cycle. He discussed several slides which had to do with a series of researches by the A. S. H. V. E. Research Laboratory relating to heat losses from the human body under various atmospheric conditions and the proportion of heat that is lost by evaporation, radiation and conduction, and how these factors vary with the change of temperature. Mr. Carrier showed how for every individual normally clothed there is a certain optimum effective temperature at which he experiences the greatest comfort both in summer and in winter. He mentioned the value of scientific air conditioning in promoting human comfort and efficiency by maintaining these conditions

and gave examples of equipment designed for several large and notable installations. The need for research on methods of discharging and distributing the air in occupied places was graphically shown by the speaker. It was also pointed out that a great part of a person's time is spent in the home where most can be done to conserve the health and comfort of the occupants, therefore, extremes of temperature and excessive dryness should be prevented. The zenith of present methods has been approached but the demand for human comfort will overshadow considerations of cost, in Mr. Carrier's opinion so that some day the average householder will be able to produce in his home in winter the climate which would rival that of the South and in summer time that of the seashore or mountains.

A rising vote of thanks was extended to President Carrier for his most intellectual talk.

March 9, 1931. Prof. A. C. Willard, head of the department of Mechanical Engineering at the University of Illinois, was the guest speaker of the evening. His subject was Ventilation Investigation for the New Chicago Subway, which was well received by the 150 members and guests in attendance.

Professor Willard's description of the experiments conducted, his thorough knowledge of the problems involved, and his discussion of the proposed system of ventilating the Chicago Subway, all contributed to make a most successful and interesting meeting. The chapter extended a "rousing" vote of thanks to Professor Willard, and the meeting was then thrown open to discussion, in which many of the members took part.

Due to the lateness of the hour the *question box* was held over until the next meeting of the Chapter.

Massachusetts

March 28, 1931. The regular meeting of the Chapter was held in a private room at Durgin, Park & Company's restaurant at 16 Hayward Place in Boston.

Previous to the Chapter meeting the Board of Governors discussed the contents of a letter from Buerkel & Co., sent to the Chapter through President McCoy, emphasizing the necessity of using genuine wrought iron nipples on any job where wrought iron pipe is used.

President McCoy presided and appointed a committee of three; Alfred Kellogg, *Chairman*, D. S. Boyden and H. B. Wiegner, to nominate two members for the Board of Governors, the same to be elected by ballot at our April 21st meeting and to fill vacancies caused by the retirement from the Board of J. W. Brinton and T. F. McCoy.

D. S. Boyden urged the members to lend their support in every way, so that the June meeting at Swampscott will be a success.

After lunch, W. F. Gilling fittingly introduced Ralph B. Wilson, Vice-President of the Babson Statistical Organization. Mr. Wilson gave a very interesting talk on statistical information regarding general business conditions with numerous digressions from statistics to drive home his point.

Minnesota

March 10, 1931. The Minnesota Chapter of the A. S. H. V. E. held its March meeting at the Men's Union Building, University of Minnesota, with 30 members and guests in attendance.

J. F. Hale, Chicago, the speaker of the evening, gave a brief outline of the progress made in the heating and ventilating industry since the early nineties. He stated that early engineering problems were solved by rule-of-thumb and cut-and-try methods. The industry is indebted to the Society for much of the complete engineering data available at the present time, was the thought expressed by Mr. Hale.

He went on to explain that in the ventilating field, the early forms of pipe-coil heaters have been to a large extent superseded by cast-iron heaters, and the latter is now supplemented by copper-fin heaters. A great amount of trouble will be avoided, Mr. Hale thought, if engineers and contractors will avail themselves of manfuacturers' recommendations in the installation of equipment. In the discussion that followed Professor Rowley suggested that unit-copper "radiators" be included, which idea was carried out.

Norman Downey, La Crosse, stated that the promises made for copper-fin "radiation" at its introduction about six years ago, had been fulfilled from the standpoint of initial cost, efficiency and general adaptability.

C. J. Scanlon, La Crosse, mentioned that research work with copper-concealed "radiation" showed that tables giving such factors as heat loss from buildings, condensation rates, etc., are incomplete unless they take into account the source of heat supply.

C. E. Lewis suggested that inasmuch as human comfort is affected very appreciably by humidity and air motion as well as temperature, these factors also enter into the problem.

Michigan

March 16, 1931. The Michigan Chapter of the Society held their regular monthly meeting at the Cadillac Athletic Club, with an attendance of 60 members and guests.

Pres. E. E. Dubry introduced the speaker of the evening, Dr. Lent Upson, Director of Detroit Bureau of Governmental Research and Secretary of Ralph Stone Tax Committee, whose subject was Shall We Abolish Government? Dr. Upson touched upon the tax situation, particularly in Detroit, calling attention to the fact that as science has developed, taxes likewise have of necessity increased. He so clearly explained the situation as it really exists, that perhaps the average taxpayer who listened to him will more gladly pay his next tax assessment.

This meeting marked the annual Goodfellowship Smoker and Get-Together Meeting in which the A. S. H. V. E. Banner Derby featured the program. This is the third year in which the Derby has been held, conducted by W. J. Whelan.

Western New York

March 9, 1931. The Western New York Chapter of the Society held their March Meeting, which was in the form of a smoker, at the Buffalo University Club.

Under the guidance of D. J. Mahoney and his able assistants, H. B. Roarke and W. E. Voisnet, an interesting program had been arranged. Various five-man teams were made up for bowling, which resulted in very close scores. Among the bowlers, high scores were made by Joseph Davis, Hugh Roarke, Myron Beman, John Nathan and George Shank.

The members and guests enjoyed bridge and other card games, and at intervals during the evening, other forms of entertainment were indulged in freely by all.

At 11 o'clock a very appetizing buffet supper was served, which was thoroughly enjoyed by the 55 members and guests attending.

Pacific Northwest

March 5, 1931. The members and guests of the Pacific Northwest Chapter of the Society assembled in the Gold Room of the New Washington Hotel, Seattle, where they held their March meeting.

Following the usual dinner the meeting was called to order and the minutes of the previous meeting were read and approved.

W. L. Dudley reported on the proposed license law for engineers, which was followed by the appointment of the Nominating Committee by Prof. E. O. Eastwood, as follows: W. L. Dudley, *chairman;* E. L. Weber and C. Zokelt.

R. S. Whaley, of the Power Plant Engineering Co., was the guest speaker of the evening. Mr. Whaley gave a very interesting talk on Oil Burners, which brought about considerable discussion.

February 4, 1931. As customary the members and guests met at the New Washington Hotel for the February meeting of the Chapter, which was presided over by E. O. Eastwood.

After the usual order of business Professor Eastwood reported on his trip to Pittsburgh, where he attended the Annual Meeting. He later introduced the guest speaker of the evening, F. C. Houghten, Director of the Society's Research Laboratory in Pittsburgh, who rendered a very enlightening and detailed report of the activities at the Laboratory.

Mr. Houghten's talk was most interesting as it gave the members of the chapter an opportunity to learn of the work going on at the Society's Laboratory. His visit was very much appreciated by the Chapter members.

January 5, 1931. The January meeting of the Pacific Northwest Chapter was held in the New Washington Hotel, Seattle.

An interesting program had been preparel by the Program Committee, and the speakers included Mr. Albue, of the Liquid Carbonic Gas Co., who explained the manufacture and use of dry ice; J. D. Ross, of the City Light Co., who gave a demonstration and explanation of the transmission of electricity without the use of a conductor; Mr. Pollard, of the Gas Co., who spoke on the use of gas for heating and Mr. Hoyt, of the National Gas Co., who discussed the future of the gas industry.

Following the election of the representative and alternate to represent the Chapter at the Annual Meeting, the meeting was adjourned.

Philadelphia

April 9, 1931. The Philadelphia members and guests met at the Engineers' Club for their regular monthly meeting, which was called to order by Pres. E. N. Sanbern.

M. F. Blankin, Chairman of the Meetings Committee, introduced Commander E. L. Gayhart, Superintending Constructor of the U. S. Navy at the New York Shipbuilding Corp., who presented a very interesting paper on the requirements for heating and ventilating of Navy ships.

At the conclusion of Commander Gayhart's talk, Mr. Blankin introduced W. T. Breckenridge, who is in charge of the heating and ventilating on Government ships in the New York Shipbuilding Corp. Mr. Breckenridge also gave a very interesting talk on the methods and problems of ship heating and ventilating.

The Treasurer's report was then submitted and approved as read, and was followed by a request from Mr. Nesbitt, chairman of the Membership Committee, for the co-operation of the entire Philadelphia Chapter in order to increase the present membership.

Announcement was made that the May meeting of the Chapter, which will be held on the 14th, will consist of the annual outing to be held at Tavistock Country Club, followed by the regular meeting in the evening, and a tour of inspection of the plant of Warren-Webster & Co., Camden, N. J.

Those elected to serve on the Nominating Committee for the ensuing year are, H. G. Black, F. D. Mensing, J. D. Cassell, R. C. Bolsinger and A. McClintock, Jr.

Mr. Mensing spoke briefly regarding the summer meeting to be held at Swampscott, Mass., and requested as many as possible to attend.

March 12, 1931. The March meeting of the Philadelphia Chapter, which was held at the Engineers' Club, was called to order by Vice-President Davidson.

Edgar Oters and Joseph Doerr, both of the Engineers' Club, spoke in behalf of the Club, requesting that as many members as possible of the Society take advantage of the many facilities offered by the Club.

R. V. Frost was then presented by the Chairman of the Meetings Committee, M. F. Blankin. Mr. Frost gave a very interesting talk illustrated by lantern slides on the Combustibility of Coal.

At the conclusion of Mr. Frost's talk, M. C. Gillett conducted a general discussion with blackboard illustrations of several problems on low pressure steam and hot water heating.

Mr. Blankin announced that the subject for next meeting of the Chapter would be Methods of Heating and Ventilating in Ship Construction.

Pittsburgh

March 9, 1931. The regular monthly meeting of the Pittsburgh Chapter was held in the Assembly Room of the Fort Pitt Hotel, Pittsburgh, with an attendance of 39 members and guests.

Following the reading of the minutes of the previous meeting, which were approved as read, F. C. McIntosh presented the final report of H. Lee Moore, General Chairman of the Committee on Arrangements for the Annual Meeting. In this report, appreciation was expressed for the wholehearted support and co-operation which was given by the members of the committee and by the members of the local Chapter, which made the Annual Meeting such a success.

In accordance with suggestions made by the Committee on Arrangements, it was moved, seconded and unanimously carried, that the balance of the money in the possession of the Committee on Arrangements be disposed of by placing one thousand dollars in a savings account for this Chapter, with the understanding that all interest be added to the principal for use at some future meeting, and the remaining amount be placed in the general fund of the Chapter to reimburse it for expenses incurred at the meeting.

F. A. Gunther, reporting for the Program Committee, announced that V. S. Day, Carrier-Lyle Corp., would speak at the April meeting on warm air heating, with special reference to the newly-developed gas-fired fan-circulation installations.

F. C. Houghten introduced Prof. T. G. Estep, of the Engineering Department of Carnegie Institute of Technology, who gave a very interesting talk on the Small Stoker for Domestic Heating Plants. Prof. Estep reviewed parts of the Fuels Section Meeting of the *A. S. M. E.*, held recently in Chicago, and outlined the requirements for the small stoker, listing in order of their importance: reliability; adaptability to any type of coal; low first cost; low maintenance cost; minimum noise; no blow-by of gases through the stoker hopper; low power consumption; no arching of coal in the hopper; and ability to remove ash without raising dust.

He also spoke of the possibility of selling heating service, either on a measured or a flat rate basis, and of the possibility that the smoke nuisance may compel the universal use of stokers.

After many interesting questions had been raised, and discussions given, a rising vote of thanks was given to Professor Estep.

St. Louis

April 1, 1931. The St. Louis Chapter meeting for April was held at the Roosevelt Hotel, where 39 members and guests assembled for the evening.

It was announced by Pres. R. M. Rosebrough that W. H. Carrier, president of the Society, would be in St. Louis on April 15, when he would address the Electrical Board of Trade.

The meeting was then turned over to L. Walter Moon, chairman of the Program Committee, who introduced E. A. Jones, of the Mueller Furnace Co., Milwaukee. Mr. Jones, who is a charter member of the St. Louis Chapter, very ably discussed the question of gas heating in homes, with particular reference to the change from manufactured to natural gas and its effect upon various types of burner appliances. Mr. Jones further described the application of gas heating in connection with air conditioning as well as the increasing demand for this type of apparatus. After considerable discussion by the members, a rising vote of thanks was extended to Mr. Jones for the manner in which he handled his subject.

Mr. Moon presented several questions received in the Question Box.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 14 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by May 15, 1931, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ASHLEY, CARLYLE MARTIN, Engineer, Carrier Engrg. Corp., Newark, N. J.	W. H. Carrier D. E. French	A. E. Stacey, Jr. J. I. Lyle
BROWN, THOMAS, Contracting Engineer, Own Business, Toronto, Ontario, Canada.	A. W. Givin R. J. Millar	William Philip J. J. MacKenzie
BURTON, W. DEAN, Chief Engr., Combustioneer, Inc., Goshen, Ind.	H. R. Linn H. G. Kreissl	C. W. DeLand H. G. Thomas
CROSS, ROBERT EARL, Sales Engr., Minneapolis-Honeywell Regulator Co., Boston, Mass.	E. A. Dusossoit Edgar Shaw	J. S. Webb Alfred Kellogg
CUNNINGHAM, TOM M., Branch Manager, Carrier Engineering Corp., Dallas, Texas. (*Advancement*)	M. M. Smith L. L. Lewis	E. T. Murphy J. I. Lyle
DUBOIS, LOUIS J., York Ice Machinery Corp., York, Penna.	J. H. Hucker J. R. Hertzler	H. P. Waechter S. E. Plewes
HOLLISTER, EDMUND WALLACE, Field Supt., New York University, New York, N. Y.	R. D. Morrill A. C. Giannini	F. K. Teichmann (*ASME*) A. I. Peterson (*ASME*)
HATTON, ALBERT E., Mech. Supt., State of Michigan, Detroit, Mich.	J. R. McColl J. W. Snyder	Dermid McLean L. G. Parrott
JAMES, HAMILTON ROYCE, Industrial Engr., United Engineers & Constructors, Inc., Philadelphia, Penna.	Pierce Timmis L. C. Davidson	C. E. Mayette C. S. Wilmot
REDSTONE, ARTHUR L., Research Engr., Proctor & Schwartz, Philadelphia, Penna.	Herman Bogarty M. Sheffler	Samuel Gross
RHEA, CHESTER A., Heggie-Simplex Boiler Co., Philadelphia, Penna.	W. H. Wild M. F. Blankin	C. V. Haynes M. C. Gillett
SMITH, GEORGE G., Board of Education, Chicago, Ill.	John Howatt J. J. Finan	J. P. Fleming E. J. Finan
WHALEY, RALPH S., President, Power Plant Engrg. Co., Seattle, Wash.	W. L. Dudley W. W. Cox	C. F. Twist Marius Anderson
WOOD, FREDERICK C., Sales Engr., York Ice Machinery Corp., York, Penna.	J. R. Hertzler Maron Kennedy	H. P. Waechter

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

ARMSTRONG, ASHER D., Asst. Mgr., Heating Dept., Crane Company, Minneapolis, Minn.

ATKINS, THOMAS J., Standard Products Div., Carrier Engrg. Corp., Newark, N. J.

BISCH, BERNARD J., Chief Engr., St. Mary of the Woods College, Indiana.

CARR, MAURICE L., Staff Engr., Pittsburgh Testing Laboratory, Pittsburgh, Penna.

CHAPPELL, HENRY DAVID, Sales Engr., B. F. Sturtevant Co., Detroit, Mich.

CONNER, RAYMOND MOWER, Director, American Gas Assn. Laboratory, Cleveland, Ohio.

DALY, ROBERT E., Asst. to President, American Radiator Co., New York, N. Y.

GOELZ, ARNOLD H., Vice-Pres., Brunswick-Kroeschell Co., Chicago, Ill.

GOENAGA, ROGER C., Technical Director, Ateliers Ventil, Lyon, France.

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Heating·Piping and Air Conditioning EDITORIALS

An Important Transportation System

Mass transportation depends on a steady flow for volume. One-way streets, synchronized traffic, vehicular tunnels, escalators, etc., are common examples; while the mine, factory, warehouse, construction job, etc., all rely more and more upon conveyors of some kind to expedite the handling of materials and reduce time and labor.

The oldest and most widely used form of continuous conveyor is the pipe line, suitable, of course, only for materials which can be made to flow. Pipe lines are essential to our modern complex life and we are served with steam, gas, oil, water and refrigeration by great systems of pipes from far distant sources or from central stations.

One needs only to observe the construction of a New York subway to realize somewhat how full of pipes is the earth beneath a great city. Like veins and arteries, they are the unseen servants of man, upon which his health and safety rest. In time of war or earthquake they are the most vulnerable spots. When a great artery is broken, the situation instantly becomes desperate. Piping in industrial plants and other buildings is no less important; failure of a piping system may paralyze the entire organization.

Great progress has been made in piping and new and more difficult problems are constantly being solved. Longer and larger pipe lines are being built; higher pressures are being carried; more durable materials are being developed, and, through research and experience, reliability in operation is continually better assured.

Likewise, pipes can now be used to convey materials formerly considered unsuited for such handling. Many chemicals, fluid enough to flow readily but of a corrosive nature, can now be piped satisfactorily because new and more resistant materials are available for the construction of pipes, valves and containers. Highly irritant, poisonous and inflammable liquids can be manufactured and handled with more ease and safety in fully closed systems. Food products can be kept free from contamination and be handled more rapidly.

The piping engineer is one of the key men in our industrial plants, our cities, our big hotels, hospitals and other structures. It is a part of his responsibility to keep in close touch with the engineering progress made in this important subject.

Spring Is Here!

"I like Spring; it always comes at such a nice time of the year." Many of us feel that way about these few weeks when the weather is just right for work, recreation or sleep.

Spring as a season has received a lot of credit from poets, painters and writers since the world began. Of course, it is the Spring weather that they've really meant; days not too warm nor yet too cold—refreshing atmosphere at night—air neither too dry nor humid.

Many people are fortunate enough to benefit by the ideal atmosphere of an ideal Spring day (there aren't so many, by the way) throughout the year. They are the ones who are enjoying real air conditioning. Industrial plant managers are finding new advantages in the control of atmospheric conditions, both to the manufacturing process and for the comfort and health of the employes. Owners of theaters, restaurants and hotels know that a refreshing atmosphere is essential to business success.

While no poet may ever write, "In an air conditioned building, a young man's fancy lightly turns to thoughts of love," proper air conditioning can give us ideal Spring weather no matter what the season. Realization of this is more widespread every day; the prospects are bright.

No Faucets Leaking

We've been through, around and under more than several large electric generating plants, of course, but we never fail to experience a purely emotional reaction, aside from any technical interest involved. There's a thrill (for us at least) in a modern turbine room and a keen realization of the attention to detail that must go into the design, the installation and the operation of the whole plant.

Last week, the chief operating engineer of a plant that has an enviable reputation for efficient production of energy showed us around. Starting at the coal dumper, we ended up where experiments were being made on the utilization of the ash. Then we followed

the steam from the boilers to the turbines, looked over the control room, and finished at the switch-yard.

Stepping into a wash-room to clean up (we'd been on the roof as part of the tour) our guide commented: "You'll notice none of the hot water faucets are leaking. The only way to make money in this business is to watch every detail."

We think it a commendable position to be in to operate a plant of such magnitude carefully enough so that one or two leaking faucets in a wash-room would show up in the costs. Many an industrial plant engineer is in the same position; those who are not should realize more fully the advantages that can be had by careful attention to the smallest detail.

A Larger Group Needed

The early years of air conditioning were characterized by a moderate growth, carried on by a comparatively small number of engineers, interested primarily in developing essential fundamental knowledge regarding the phenomena involved. These men were, almost universally, well-trained, competent engineers. With few installations from which experimental data could be had, each new job was based almost directly on the fundamentals.

During the past few years—three or four, say—air conditioning as a branch of engineering that affects almost every industry or business activity has "blossomed out." More men have been needed in the designing, the installing, and the operating departments. Most of these men have had the advantage of careful instruction and training under the "pioneering group."

This recent, more rapid, growth in the air conditioning industry has been due to a realization on the part of industrial executives, business men, and the public that the control of atmospheric conditions for the processing of materials or for human comfort can offer real advantages and effect large economies. Air conditioning information has been broadcast in general advertising, and through technical journals, including this one. A spread of engineering information—as distinguished from general information—has been inherent to this growth. Successful performance of the earlier air conditioning plants has, of course, been of paramount importance to this more rapid development.

And now all signs point to an even more rapid development in the next few years. Many look at air conditioning as the most promising industry of the future—one that will show the same aggressiveness that the automobile industry, for instance, showed in its years of growing importance. This future growth of air conditioning depends upon a much larger body of men trained in the fundamentals as carefully as were the "pioneering group" to which air conditioning owes its foundation. These engineers so necessary to air conditioning during the next few years can be given the information they need through direct training and through the presentation of carefully prepared, reliable, technical articles in the engineering press.

It is to be hoped that the growth of air conditioning will continue to be an engineering achievement, rather than one based on undue sales emphasis at the expense of technical development. Success of every installation will maintain the reputation the industry now enjoys; such success can be insured by a large enough body of engineers well-versed in the fundamental knowledge inherent to the art. Every engineer concerned with the design, the installation or the operation of air conditioning equipment must have the benefit of reliable information on all phases of air conditioning if development is to be as great as expected.

Next Month—the Giant Humidor: Complete insulation of the central plant of Bayuk Cigars makes close control of temperature and humidity possible. Were it not for the special construction, size of the air conditioning plant would be increased 30 per cent, which would have made its cost prohibitive. R. D. Touton tells—in detail—how the plant is constructed; considers the advantages of insulation; points out the results obtained in the June HEATING, PIPING AND AIR CONDITIONING. When Bayuk cigars leave the "Giant Humidor" they are in perfect smoking condition, thanks to air conditioning aided by proper insulation.

Industrial Plant Engineers can find a wealth of ideas in modern steam power plants which can be applied to industrial piping problems. Plant piping presented one of the major problems in the design and construction of Plant Atkinson, a 60,000 kw installation near Atlanta, Georgia. For this reason E. C. Gaston's article in next month's number is devoted entirely to the piping and accessory equipment, including a discussion of the problems encountered in the design, considering the reasons for the selection of equipment, and the specifications for the piping systems.

Some Reports on Research

M. M. Davidson, Building Manager, Studies His Air Conditioning Plant Continually to Better its Performance

Instruments Necessary to Control Air Conditioning

"THREE things are especially to be noted in our final layout," states M. M. Davidson, manager of the Northern Trust Co. building, Chicago, which was described in HEATING, PIPING AND AIR CONDITIONING last month. "All exposed sides of the building have been treated with double windows. The large window expanse in buildings of this type cools air quickly, and the resulting constant slight movement of air about high windows gives an impression of draft. The gain made in comfortable work space close up to windows in office buildings will justify the installation of double windows.

"Air conditioning, whatever the system, is only as good as air distribution. We have a large number of distribution points for incoming air within the rooms, with inlets at the ceiling. Special lighting fixtures have been installed beneath ornamental grille openings that appear to be a part of the lamp itself. The top of the bowl is a flat glass disc which deflects the incoming current of air. Diffusion is favored and the air comes down gradually over a widely distributed area. Forced exhaust is at the floor line. The air inlets are in side wall positions in some of the earlier sections of the building, but the principle of multiple diffusion units holds throughout. This is especially important when volumes of cooled air pass through the system in summer. It prevents noise.

"The third factor, and an indispensable tool in precision handling of the plant, is the multiple installation made of resistance thermometers. There are twenty-two dry cell resistance thermometers in the system, all connected on a low tension electric circuit and registering on a compact indicating system at my desk. These thermometers are located as follows:

"One in the spray chamber of each of the six air conditioning units, to indicate the saturation point of tempered air supplies. Indirectly, they indicate the relative humidity of the several floors.

"There is a nickel resistance thermometer in each fan discharge, an index of temperatures of air discharged to the several floors.

"Resistance thermometers are placed at strategic positions on walls or columns of every floor. These show the final room temperatures. Outside wet and dry bulb readings are also taken. Continuous charts are kept of readings taken daily at ten, twelve, and at three of (a) dew point readings in the spray chambers; (b) of fan discharge temperatures; and (c) temperatures on the various floors.

"Remote control is not employed, but the central index is so delicate and so well understood in terms of the building needs that we know at once what has failed if temperature balance has been upset at any point. A dew point reading operates two louvres. The one may lag, with the result that the room air change is not rapid enough. The central indicator enables a quick correction of such difficulties.

"We know our building. The fourth floor is windowless. The sixth floor has many windows and outside exposure on four sides and the roof. Cumulative heat from lamps and human occupancy on the fourth floor requires that tempered air come in at lower temperatures than on the other floors. The fourth floor air enters at 65 F, while the sixth floor requires air at 72 F under the same outdoor conditions. The resistance thermome-

TABLE 1—TYPICAL CHART, SHOWING READINGS THREE TIMES PER DAY OF FAN DISCHARGE TEMPERATURES, TEMPERATURE OF ALL FLOORS OF THE BUILDING, AND DEW POINT IN HUMIDIFICATION CHAMBERS AS OF JANUARY, 1931

Fan Discharge			Floor Temperature				Dew Point			
A. M. 10:00	M. 12:00	P. M. 3:00	Floors		A. M. 10:00	M. 12:00	P. M. 3:00		A. M. 10:00	M. 12:00	P. M. 3:00
11	66½	67	67 –	1	1	72	72	72 +	1	45 –	45½	46
11	66½	67	67	2	2	72	72	72 +	1	45 –	45½	46
13]	68	68	69	3	3	72 –	72	72	3	45 –	45½	43 +
12	65	65½	65½	4	4	72	72	72	2	45	46	47
13	68	68	69	5	5	72 +	72½	73	3	45 –	45½	43 +
14	71½	71 –	72 –	6-S	6	72 –	72	72 –	4	45	46	46 +
14	71½	71 –	72 –	6-N	7	72	72½	72 +	4	45	46	46 +
15	66 –	66 +	66	A	8	73	73 +	73½	5	47	46	46 +
15	66 –	66 +	66	B	9	73 +	73 +	73½	5	47	46	46 +
S. D. V 16	61	61	61 +	Out. Dry	10	23 –	27	32 –	Out. Wet 6	23 +	28	32½

Inside Humidity 38%

ters are so delicate that they could register differences of .01 F if necessary and our scales were so calibrated.

"The management routine of this building contemplates the maintenance of the desired temperature of 72 F with a relative humidity of 40 per cent during the winter control period. Supplementary heat is available through the thermostatic regulation of radiators when outside temperatures fall. The supply of tempered air is for air conditioning, but is supplemented by the heating system to adjust to lower temperatures outside.

"Without precision instruments and continuous study of a plant, it is quite possible to have complete equipment and get nothing consistently satisfactory from its operation," states Mr. Davidson. "We are continuing our study of our plant, and continue to find the way to better adjustment for good control."—*S. P. M.*

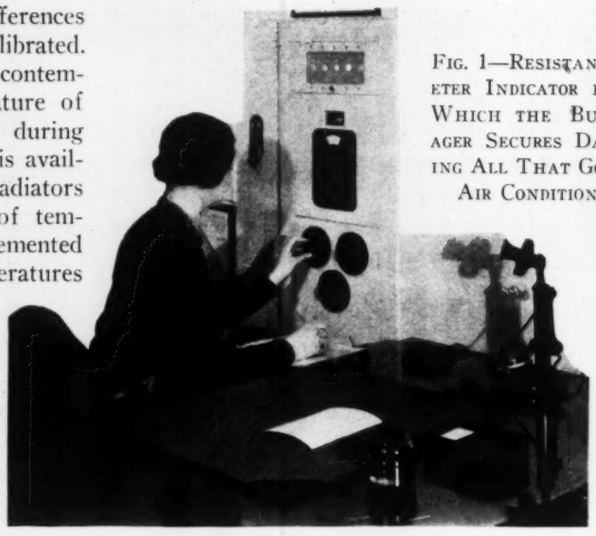


FIG. 1—RESISTANCE THERMOMETER INDICATOR BY MEANS OF WHICH THE BUILDING MANAGER SECURES DATA CONCERNING ALL THAT GOES ON IN HIS AIR CONDITIONING PLANT

TABLE 2—TYPICAL CHART OF CONDITIONS IN BUILDING DURING PERIODS OF SUMMER CONTROL

Fan Discharge				Floor Temperature					Dew Point			
	A. M. 10:00	M. 12:00	P. M. 3:00	Floors		A. M. 10:00	M. 12:00	P. M. 3:00		A. M. 10:00	M. 12:00	P. M. 3:00
11	66½	66½	65+	1	1	77½	77½	76½	1	52	52	52–
11	66½	66½	65+	2	2	76	76	76	1	52	52	52–
13	65½	65½	65½	3	3	76+	76+	76+	3	51	51	50½
12	64	64	64	4	4	75	75	75+	2	51+	51+	51
13	65½	65½	65½	5	5	74	74	74+	3	51	51	50½
14	65½	65½	65–	6-S	6	76½	76½	77	4	52	52	52–
14	65½	65½	65–	6-N	7	76½	76½	77	4	52	52	52–
15	68½	68½	68	A	8	76	76	76+	5	53	53	53–
15	68½	68½	68	B	9	76	76	76½	5	53	53	53–
S. D. V. 16	....	....	....	Out. Dry	10	84	87	90	Out. Wet 6	....	....	....
									Spray Water Temperature	39½	39½	39
									Spray Water Return	44½	39½	39

Conventions and Expositions

Heating and Piping Contractors National Association: Annual Convention, May 6-9; Kentucky Hotel, Louisville, Ky. Secretary, Joseph C. Fitts, 50 Union Square, New York City.

American Society of Refrigerating Engineers: Spring meeting, May 6-8; Hotel President, Kansas City, Mo. Secretary, David L. Fiske, 37 W. 39th st., New York City.

American Gas Association: Annual convention, natural gas department, May 11-14; Hotel Peabody, Memphis, Tenn.

National Pipe and Supplies Association: Convention, May 11-13; Hotel Hollenden, Cleveland, Ohio.

1931 Midwest Bituminous Coal Conference: May 21-22; University of Illinois, Urbana, Ill. Chairman, Program Committee, B. R. Gebhart, Illinois Coal Bureau, 307 N. Michigan ave., Chicago.

American Society of Mechanical Engineers: Meeting at Hartford, Conn., June 1-3. Secretary, Calvin Rice, 33 W. 39th St., New York City.

National District Heating Association: Annual Convention, June 2-5; Statler Hotel, Boston, Mass. Secretary, D. L. Gaskill, 603 Broadway, Greenville, O.

American Society of Heating and Ventilating Engineers: Semi-annual meeting, June 22-25; Swampscott, Mass. Secretary, A. V. Hutchinson, 51 Madison Ave., New York City.

National Association of Power Engineers: Annual convention and exposition, September 7-11; Convention Hall, Kansas City, Mo. Secretary, F. W. Raven, 417 S. Dearborn st., Chicago.

Southern Power and Machinery Show: October 19-24; Textile Hall, Greenville, South Carolina.

Third International Conference on Bituminous Coal: November 16-21; Carnegie Institute of Technology, Pittsburgh, Pa. Chairman, Thomas S. Baker, President, Carnegie Institute of Technology.

PRACTICAL PIPING PROBLEMS

Practical Hints on the Use of Meters in the Operation of Industrial Piping

by **W. H. WILSON**

Suitable Metering Devices Aid Economical Operation

Supervisors of plant maintenance work are confronted with the problem of operating the plant economically and reducing the maintenance cost for both labor and supplies. In connection with the piping equipment and its distributing lines, the use of suitable meters, gages and similar instruments will show what is being transmitted through the lines and enable the operating men to correct wasteful practices and improper working conditions.

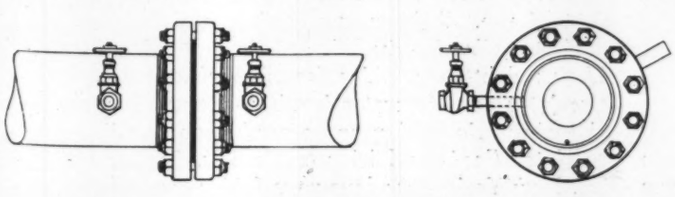
Portable Meters Useful

It is not always necessary to spend a large sum of money for an extensive amount of metering equipment. Arrangements can usually be made, by having suitable piping connections installed, so that a meter can be used in one place for part of the time and then moved to another location. By this arrangement periodical tests can be made and a close check kept on operating costs.

The flow meter can be placed alternately on the different branches of the steam lines. The compressed air supply lines can be checked in a similar manner. A recording vacuum gage placed for a few days at different places on the heating system return lines, will show if there are vacuum leaks of consequence. A check-up of the water lines, with meters, will show if more water is being used than should be necessary.

With the proper pipe connection tappings at the respective locations in the lines equipped with shutoff valves, and orifice discs in place, it is usually a simple and inexpensive matter to change the meter or gage and connect it at a new location.

It is better to have a *few* charts from recording instruments reach the operating and maintenance engineer's desk *and be carefully studied* than to have a handful brought in daily and allowed to accumulate, without any interest being directed toward them. Other things being equal, however, it would seem that the more complete the records and charts are, the more valuable they become.



FLOW METER CONNECTIONS

The Pipe Connections

The illustrations show pipe connections that can be installed, at the different locations, for the purpose of applying meters and gages.

Before installing orifice plates, nozzle tubes and pipe connections for the flow meters, it is advisable to consult the manufacturers of the flow meters regarding the locations. There are many essential points in the matter of a proper and accurate application of the equipment necessary to obtain correct readings on these meters. The manufacturers' representatives, with their knowledge and varied experience of the different applications of these measuring devices, can be of much assistance.

Recording vacuum and pressure gages can be arranged so that the screws holding the gage to the wall, support or gage board can be left at each testing location, so that when the gage is brought to the location it can be easily and quickly secured in place and connected with the union to the gage pipe. When the recording gage is removed an indicating gage, if desired, may be easily put in place of the recording gage. The indicating gage, in the absence of the recording gage, is a help to maintenance employes in securing the best results from a piping system.

When the vacuum gage is connected between the shutoff valve on the pump suction line and the pump, it is in the proper place when it is necessary to test the vacuum pump to find out if it is in proper working order. If the strainer and the shutoff valves are located so that the gage pipe can also be connected on the inlet side of the strainer, the gage may also indicate when the strainer needs cleaning. A strainer basket containing a large amount of sediment is liable to restrict the flow to the pump and while the pump may show a strong vacuum it will only be between the strainer and the pump, while the return lines may have an excessive amount of water and air that will affect the heating system. The gage, in this case, being on the inlet side of the strainer, should have a low reading and would indicate that the strainer on the suction line needed attention. A gage pipe connection at this location is advisable when the vacuum pump is located below the operating floor or where it is not readily accessible.

Testing Flow in Lines

Testing the flow through pipe lines will furnish the maintenance men with accurate knowledge of the requirements and conditions of the pipe lines.

By closing the different supply connections and closing the main shutoff valve, as shown at *B*, a